COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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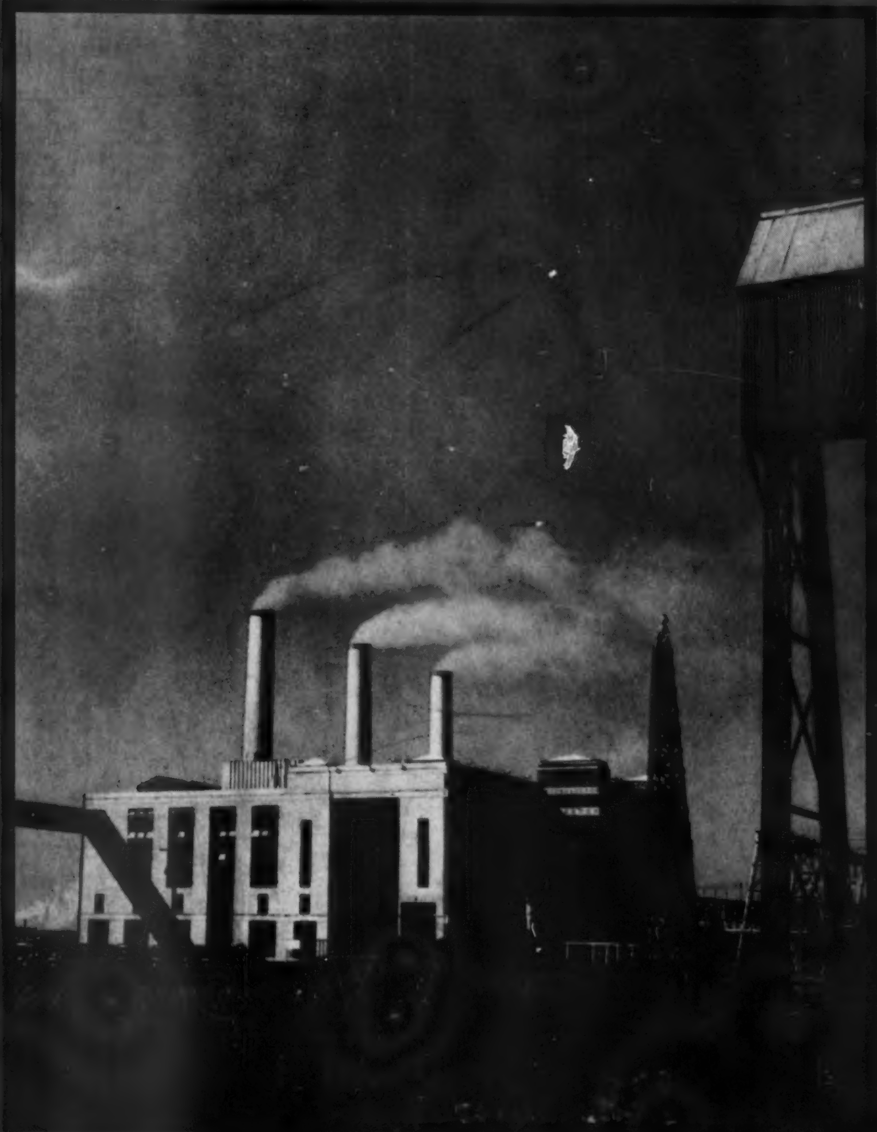

Hobart Photo Features

Steelton Plant Burns Coke Breeze and Anthracite on Traveling-Grate Stokers

Determination of Steam Quality With Throttling Calorimeter

Application of Fan Laws

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B.9

Engineering Lib.

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME TWELVE NUMBER ONE

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FOR JULY 1940

FEATURE ARTICLES

EDITORIALS

DEPARTMENTS

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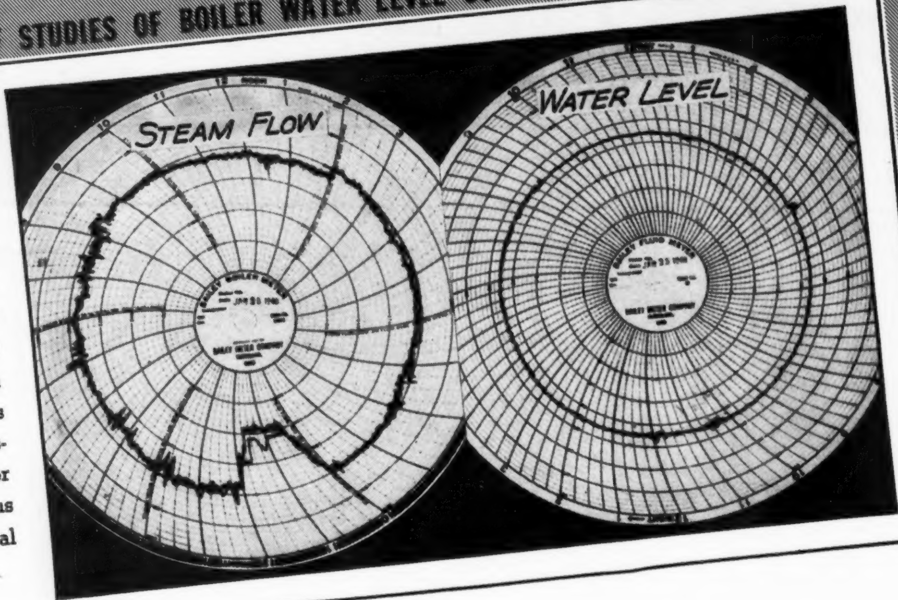

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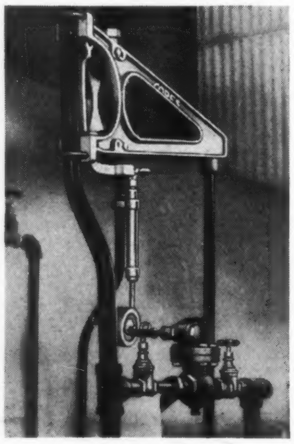

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EDITORIAL

Power Facilities

According to a statistical report issued by the Federal Power Commission in May of this year, the installed capacity of electric generating plants in this country, as of December 1939, was 40,317,924 kw. Of this, 1,275,819 kw was added last year. Scheduled additions for 1940 are given as 1,877,844 kw of which 1,332,635 kw represents private utility capacity and 545,209 kw that being installed in publicly owned plants. In addition to this the private utilities have on order or under construction 1,653,375 kw for completion in 1941.

Despite this very substantial increase in capacity and the further insistence by the utilities that they have ample reserve to meet all demands likely to be imposed by the present preparedness program, the opinion still persists in some government circles that power facilities may fall short of being adequate. Back of this contention is perhaps the desire of some individuals to see further expansion of federal power projects, or it may be the often-expressed view of the President that the present concentration along the Atlantic Seaboard of government arsenals and manufacturing plants capable of producing munitions constitutes a weakness in our defense, and that many of these plants should be located west of the Alleghanies. In the light of events abroad this reasoning is sound and it will probably govern the selection of sites for proposed government munitions plants, as well as the allocation of certain orders to manufacturing establishments.

The extent to which such dispersion of munitions manufacturing capacity may affect power supply is a matter now receiving consideration, according to reports emanating from Washington.

Increased Enrollment in Engineering Schools

The Progress Report of the Society for the Promotion of Engineering Education, as presented at the Annual Meeting in Berkeley, Calif., June 24 to 28, reveals a substantial increase in enrollment in engineering schools in the United States and Canada. Statistics covering approximately one hundred and fifty such schools show 106,000 undergraduate students, which represents an increase of nearly thirty per cent during the last three years. Moreover, candidates for advanced degrees have more than doubled during this period. While mechanical engineering shows the greatest increase, namely eighty-nine per cent, there is also a marked trend toward chemical engineering, which is a field that offers diverse and substantial opportunities for expansion in the development and production of new products.

Compared with enrollment figures a few years back, during the midst of the depression, this is an encouraging and timely comeback, especially in view of the demands that industry is certain to make upon the engineering profession, not only in the present emergency but also in meeting future competition from totalitarian countries, in which it will be necessary to pit ingenuity and mass production against lower labor standards prevailing abroad.

The increase in candidates for advanced degrees augurs well for research which has frequently been emphasized as the basis of new industrial frontiers to absorb unemployment.

Coal and Preparedness

As mentioned by one of the speakers at the recent Appalachian Coals Engineering Conference in Washington, industrial steam plants throughout the country represent a total coal consumption of about three times that of the electric utilities. The latter, while increasing their output by fuel more than threefold during the last twenty years, have increased their fuel consumption barely thirty per cent, due to improvements in design and operating efficiency. Unfortunately, no comparable figures covering industrial plants as a whole for this period are available. Although the last few years have seen very substantial improvements in industrial power plant practice, with the adoption of advanced steam conditions and through employment of more efficient steam-generating and fuel-burning equipment, there are still many plants that have yet to be modernized and which, in the aggregate, make the industrial coal consumption greater than necessary.

Coal is indispensable to industry and, as many will recall, its availability and transportation became a vital factor in the prosecution of the World War. Every ton of coal saved at the point of consumption means not only less to be shipped but also a large saving in the fuel required for its transportation.

Present indications do not warrant the assumption that there would be a repetition of the 1918 experience in connection with our vast preparedness program, inasmuch as the railroads are now much better able to handle greatly increased freight traffic and, due to a decrease in domestic consumption with the increased use of competitive fuels, excess capacity is available at the mines.

Nevertheless, coal is certain to play an important rôle in our preparedness, and those industrial establishments whose managements have had the foresight to put their power plants in order, will be in a more advantageous position to meet conditions arising from emergency orders, not only as concerns availability to handle continuous and heavy load demands but also rising coal prices and unforeseen interruptions in fuel supply due to possible labor troubles.

Steelton Plant Burns Coke Breeze and Anthracite on Traveling-Grate Stokers

By H. K. MILLER, Combustion Engr.
Steelton Plant, Bethlehem Steel Company

In laying out this installation careful study indicated that the available coke breeze and river anthracite could be best burned together on traveling-grate stokers with the anthracite forming the lower and the coke breeze the upper layer. Blast-furnace gas is also burned up to the limit of its availability and the speed of the stoker is automatically controlled to make up the deficiency. Initial operation indicates an efficiency of 79.6 per cent with this arrangement, which measures well up to the anticipated performance.

INITIAL operating experience with the traveling-grate stokers which are a feature of the new central boiler plant at Bethlehem Steel Company's Steelton Plant, Steelton, Pa., indicates an efficiency that measures well up to the anticipated performance. The stokers are used for firing anthracite screenings and coke breeze which supplement blast-furnace gas. They were chosen in preference to pulverizers as offering the most satisfactory method of burning the available fuel. Records for March show an average efficiency of 79.6 per cent.

A favorable circumstance to this performance, however, was a consistently high operating rate and a steam demand close to the plant's capacity for most of the period. Nevertheless, the showing is considered to be most satisfactory and is held to be partly accounted for by the method employed in combining the fuels and the efficient operation of automatic controls that prorate consumption according to availability of the three fuels.

In the month of March the fuel burned was 3932 tons of river coal, 2101 tons of coke breeze and 676,692,000 cu ft of blast-furnace gas. Coal and coke breeze were burned together and gas up to its limit of availability.

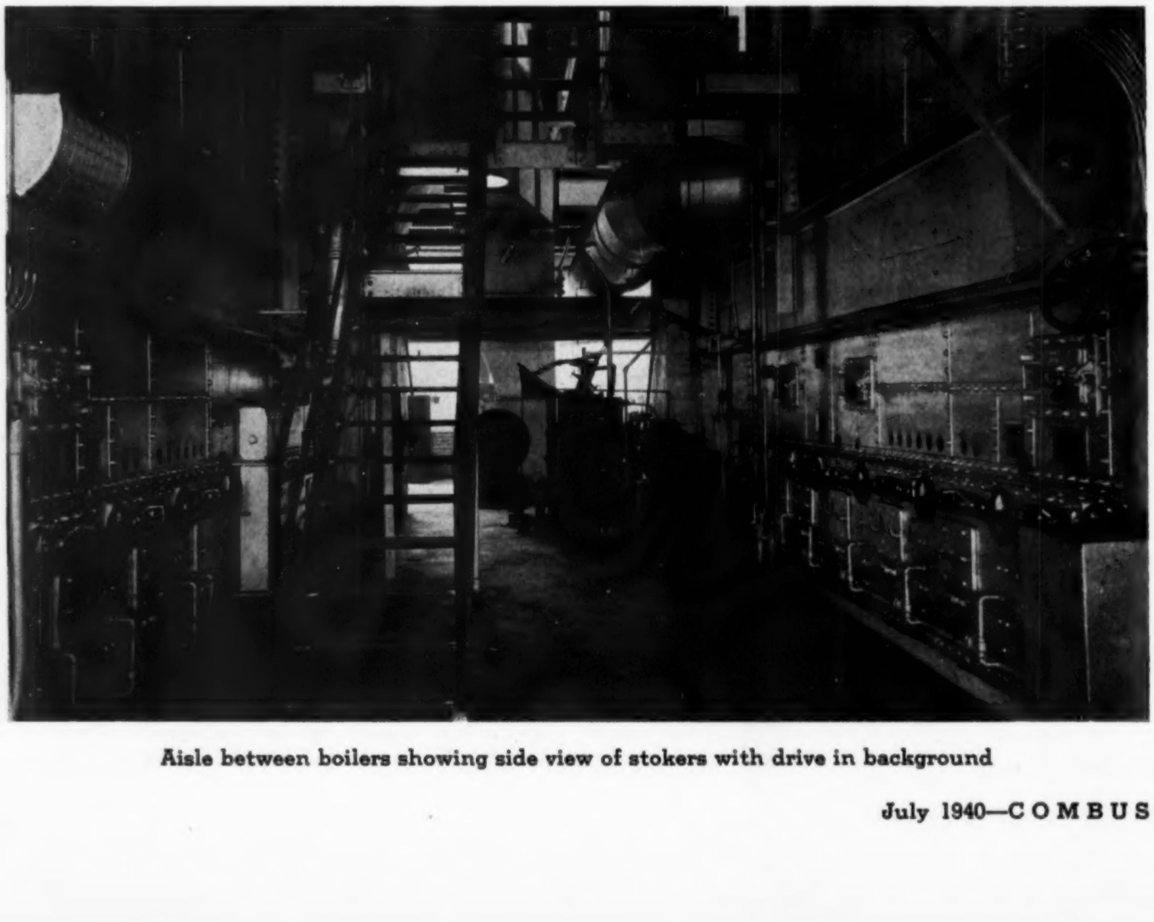
FUEL ANALYSES

	Volatile, Per cent	Fixed Carbon, Per cent	Ash, Per cent	Sulphur, Per cent	Moisture, Per cent
River coal	8.25	77.10	14.65	0.52	9.65
Coke breeze	5.40	74.45	20.15	1.80	14.0

	CO_2, Per cent	O_2, Per cent	CO, Per cent	H_2, Per cent	CH_4, Per cent	N_2, Per cent
Blast-furnace gas	11.2	0.1	26.8	2.9	0.6	58.4

In the coal analyses shown, the volatile, fixed carbon and ash total 100 per cent. These determinations are made on the dry coal basis. Most of the sulphur is included in the volatile matter.

The total steam production for the month was 149,630,000 lb with an average load of 201,000 lb per hr, or



Aisle between boilers showing side view of stokers with drive in background

67,000 lb per boiler. The maximum hourly output was 395,000 lb with peaks of over 450,000 lb, or 150,000 lb per boiler. Under these conditions of high overload the average weighted efficiency would be approximately 75 per cent.

Following is a tabulation of hourly operating data for average loads of 70,000 and 100,000 lb of steam per hour per boiler when burning coal and coke only and when burning blast-furnace gas in conjunction with coal and coke.

HOURLY OPERATING DATA

	Coal and Coke	Coal and Coke	Coal, Coke and BF Gas
Pounds of steam per hour	70,000	100,000	100,000
Steam pressure leaving superheater, lb per sq in.	250	250	250
Temperatures, deg F			
Steam leaving superheater	495	500	510
Feedwater to economizer	210	210	210
Feedwater leaving economizer	225	225	235
Gas at boiler outlet	510	525	550
Gas leaving air heater	365	375	425
Air to stokers	300	300	320
Air to gas burners	...	...	340
CO_2 leaving economizer, per cent	13–14	13–14	17–19
Air pressure in wind box, in. H_2O	4	5	2.50
Furnace draft, in. H_2O	−0.10	−0.12	±0.05
Draft losses, in. H_2O			
Boiler and superheater	0.50	0.60	1.00
Economizer	2.50	2.60	3.50
Air heater	1.00	1.00	1.20
Total	4.00	4.20	5.70

This new central boiler plant, which will effect a large saving in the cost of steam generation, replaces five smaller plants, some of which will be maintained for use in emergency. The company is also retaining the waste heat boilers on open-hearth furnaces which are an important source of steam. The steam generating equipment of the plant consists of 3 three-drum bent-tube boilers each rated at 125,000 lb of steam per hour and operated at 250 lb pressure and 475 F at the superheater outlet. The economizers are of the continuous-loop type and the air preheaters of the tubular type. Each unit is fired by a 453-sq ft traveling-grate stoker of the bar-and-key type, together with two pre-mixing blast-furnace gas burners of 20,000 cfm capacity. The furnaces have water walls front and rear and over the front and rear arches; the side walls are also water cooled from the grate to the bottom drum. With a furnace volume of 7462 cu ft a maximum heat release of about 20,000 Btu per cu ft per hr is provided. The forced-draft fan for each boiler has a capacity of 50,000 cfm of 80-F air at 8-in. static pressure and the induced-draft fan has a capacity of 100,000 cfm of 450-F air at 12-in. static pressure; both being turbine-driven through reduction gears. Automatic combustion control is employed. A 400,000-lb per hr hot-process feedwater treating system, including jet heater, deaerating atomizer and vent condenser, is installed, with dry chemical feeders for the lime and soda and solution tanks for the sulphite and phosphate. The phosphate feeders have automatic control.

To meet the steam demands at the Steelton plant it is necessary to pipe the steam nearly two miles from the West End Plant, where the main steel-making division and the new boiler plant are located, to the East End Plant where some manufacturing operations are performed. It has been found, however, that the economy of operating a single boiler plant, using blast-furnace gas and coke breeze as the principal fuels, over the operation of several smaller plants, more than justifies the expenditure for the two-mile pipe line with its inherent heat losses and pressure drop.

Three 3500-kw gas-engine-driven generators operating on blast-furnace gas and one 4500-kw steam turbine-generator supply the electricity for operation of the entire plant. There is a 7000-kw generating station supplied with steam from the general steam system for use when the gas engines are not available.

The steam is used for operation of gas producers, steam-hydraulic presses and hammers and other miscellaneous process and heating requirements. About one-third of the electric power is supplied by the steam-turbine-driven unit.

Blast-furnace gas and coke breeze are by-products of operations at the plant and the anthracite is obtained by dredging and washing the screenings which are carried down the Susquehanna River from the mining districts up stream.

The existing turbine-generator and other steam-driven equipment in the steel division provided the upper limit of pressure and temperature that could be considered for the new plant. The lower limit, of course, was determined by the pressure and temperature necessary at the end of the two miles of steam piping.

Anthracite screenings and coke breeze are fed separately to the hoppers in such a way that they feed to the stoker grates in two layers, anthracite on the bottom and coke breeze above. The baffle plate in the hopper is adjustable so that the relative thickness of the two layers of fuel can be readily varied.

Dust and Cinders Reburned

Dust reclamation has effected a considerable improvement in operating efficiency. It is accomplished by the use of a steam-jet-vacuum removal system to take the material from the hoppers beneath the economizers, air preheaters and rear boiler pass, and deliver it over the stoker grates.

Blast-furnace gas is burned to the limit of its availability and the boiler control system automatically makes up the deficiency with anthracite screenings and coke breeze by varying the speed of the traveling grates. The amount of air for combustion and temperature of air fed to the traveling grates is automatically regulated by this system.

A large proportion of the raw feedwater is provided by the gas-engine cooling water which reaches the feedwater heater at about 140 F. Exhaust from the steam-turbine-driven induced draft fan and four pumps is used to heat the feedwater. This condensate is returned to the system, but the balance of the boiler feed is makeup because the wide distribution of the consuming units makes it impractical to provide return lines.

The problem of treating the feedwater is difficult because of the large per cent of makeup and the variations in the Susquehanna River water. The river water is fed into a hot-process lime-soda softener and then through pressure filters to the boiler feed pumps. Sodium sulphite is pumped continuously into the suction of the boiler feed pumps in varying amounts according to the requirements shown by frequent sampling. Di-sodium phosphate is fed directly to the steam drum of each boiler.

Designing consulting engineers on this installation were United Engineers and Constructors, Inc., of Philadelphia. Sheppard T. Powell of Baltimore was the consulting chemical engineer in the solution of the feedwater problem.

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Determination of Steam Quality With Throttling Calorimeter

Although the throttling calorimeter has definite limitations it is still a valuable test instrument when its results are correlated with data on operating conditions. The author discusses principles, design, limitations and calculations involved in steam calorimetry and gives practical suggestions as to its application to steam boiler testing.

By P. B. PLACE
Combustion Engineering Company, Inc.

SATURATED steam delivered from the drums of commercial boilers usually contains a small amount of unevaporated boiler water mechanically entrained. This moisture carryover, as it is termed, contains boiler water salts in solution and in suspension which may be the source of deposits in the superheater or turbine when the carrier moisture is evaporated. An excessive amount of moisture carryover not only is a source of solid deposits but causes reduction in superheat and may contribute to corrosion in superheaters.

The term "Steam Quality" is used in referring to the wetness of a saturated steam and it may be expressed in terms of per cent moisture or per cent dry steam, *by weight* in the wet mixture. The amount of moisture varies from less than 0.25 per cent for a commercially dry steam to several per cent under priming or foaming conditions. It is customary to guarantee that a boiler will deliver steam having not more than 0.5 per cent entrained moisture.

Steam quality is best determined by a throttling calorimeter. The method is well established and is covered in the A.S.M.E. Power Test Codes. The principles involved are relatively simple yet care is needed to obtain reliable results.

The calorimeter is simply a device in which a flowing sample of saturated steam of known temperature and pressure is expanded through an orifice to a lower pressure that is usually atmospheric. The passage through the orifice and calorimeter involves no loss in heat other than radiation losses and since the total heat in the sample is greater than that of saturated steam at atmospheric pressure, the excess heat in the sample is used up in superheating the expanded steam and/or evaporating any moisture in the sample.

The amount of excess heat in the sample, and therefore the amount of superheating and/or the amount of moisture that can be evaporated in the calorimeter, varies with the operating pressure. The capacity of a calorimeter varies with pressure. In Fig. 1 is shown the maximum moisture that can be determined by an atmospheric exhaust calorimeter at various pressures. Fig. 2 shows the maximum temperature of the exhaust steam at the calorimeter outlet for moisture contents up to 1 per cent and different inlet pressures. Steam having over one per cent moisture is usually unsatisfactory and is difficult to sample.

Above and below 450 lb pressure, the total heat in saturated steam decreases and the excess heat and calorimeter capacity decreases. The practical operating range for throttling calorimeters exhausting to atmosphere is from 150 to 900 lb. Above 900 lb the radiation losses from high-pressure valves and fittings become excessive.

The apparatus required for test consists of a suitable means of sampling the steam, the calorimeter, and means

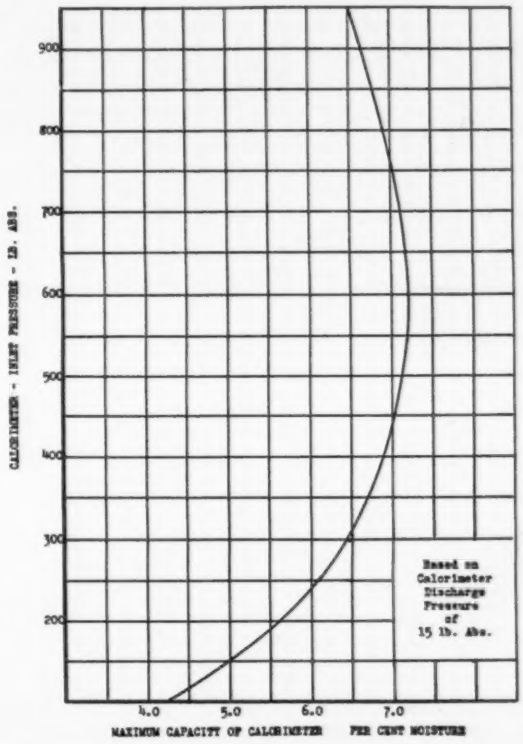


Fig. 1—Maximum moisture determinable at various inlet pressures

of measuring the pressure and temperature of the inlet and outlet steam.

Steam sampling is not within the scope of this discussion, hence will not be dealt with in detail. Actually few data are available as to the proper method of sampling steam. It is customary to use a perforated sampling tube located in a header or in a straight run of

the main steam line. Modern boiler drums are connected to superheater headers by a number of small diameter circulator tubes rather than by a single large pipe. In such cases, although calorimeters can be attached to each circulator if desired, it is easier to sample the steam at the saturated header. Fig. 3 shows a number of methods of sampling steam in different locations.

The design of the usual perforated sampling tube is supposed to give uniform sampling across the pipe or header but since the total flow through the tube is limited by a small orifice in the calorimeter that is, in most cases, no larger than any one of the perforations in the sample tube, it is uncertain how effective the design is in giving uniform sampling. Where close checks are desired, it is best to sample at one or more places along the steam path. In some cases it is desirable to locate a vent connection at the calorimeter inlet so that the rate of flow can be varied to determine the effect on calorimeter results.

There are many designs of steam calorimeters, many of which are quite satisfactory. The basic principles involved in design are as follows:

1. The amount of flow should be great enough, and the insulation sufficient, to reduce the heat capacity of, and radiation losses from, the calorimeter to a small or negligible proportion of the total heat flow.
2. The size of the orifice should permit a maximum flow through the calorimeter without back pressure at the outlet.
3. The expansion chamber after the orifice should have sufficient volume and be long enough to allow space and time for (*a*) full expansion of the steam, (*b*) conversion of the velocity energy into equivalent heat energy and (*c*) complete evaporation of all moisture present, before the outlet temperature is measured.

Choice of the size of the calorimeter is largely a matter

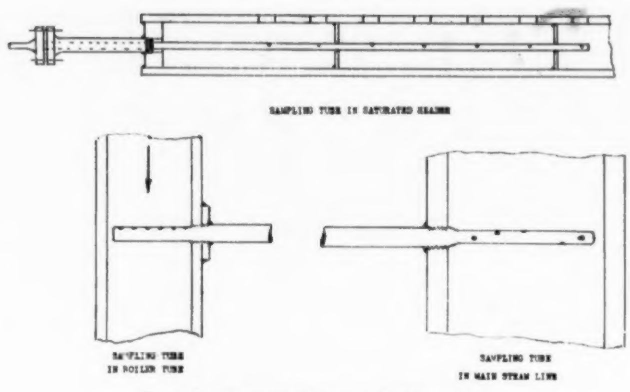


Fig. 3—Types of sampling tubes

of judgment and experience. A small-sized calorimeter having small volume and a small orifice may not give any back pressure but have considerable heat capacity and radiation surface for the amount of heat flow through it. When, however, the flow is sufficient to give low radiation losses, little is to be gained by making the calorimeter larger.

A steam flow of 250 to 300 lb per hr has been found to give satisfactory results. Allowing for a one-degree radiation loss the radiation from the insulated apparatus should not be over 150 Btu per hr. To get a flow of 250 to 300 lb per hr requires different-sized orifices at different pressures. The following tabulation gives orifice diameters that are suitable for use at different pressures.

Orifice—diam.—inches	3/16	5/32	1/8	3/32	1/16
Steam pressure range	100–200	200–300	300–500	500–700	700–850

The temperature or pressure of the inlet steam is required to determine the total heat in the sample from approved steam tables. The temperature can be measured with a calibrated thermometer or its equivalent installed in a thermometer well on the high-pressure side of the orifice. The pressure can be determined by a calibrated test gage located near the point of sampling.

Although there is no objection to the use of a thermometer well at the inlet of the calorimeter, it complicates the design and in many cases it is not necessary to measure the inlet temperature. At pressures between 400 and 500 lb small changes or errors in inlet pressure are not reflected in the total heat content of the steam. In such cases, the regular operating pressure gage is sufficiently accurate for test purposes. At pressures below 300 and above 600 lb, a change or error of 10 lb has an appreciable effect on the calorimeter results and it is desirable to measure either the temperature or the pressure of the inlet steam and, if convenient, both can be checked.

There are many designs for the calorimeter body. Some of the commonly illustrated types are shown in Fig. 4. These vary from the strictly home-made type, made of pipe fittings, to patented designs having nickel-plated finishings and special insulating features. That many of these designs

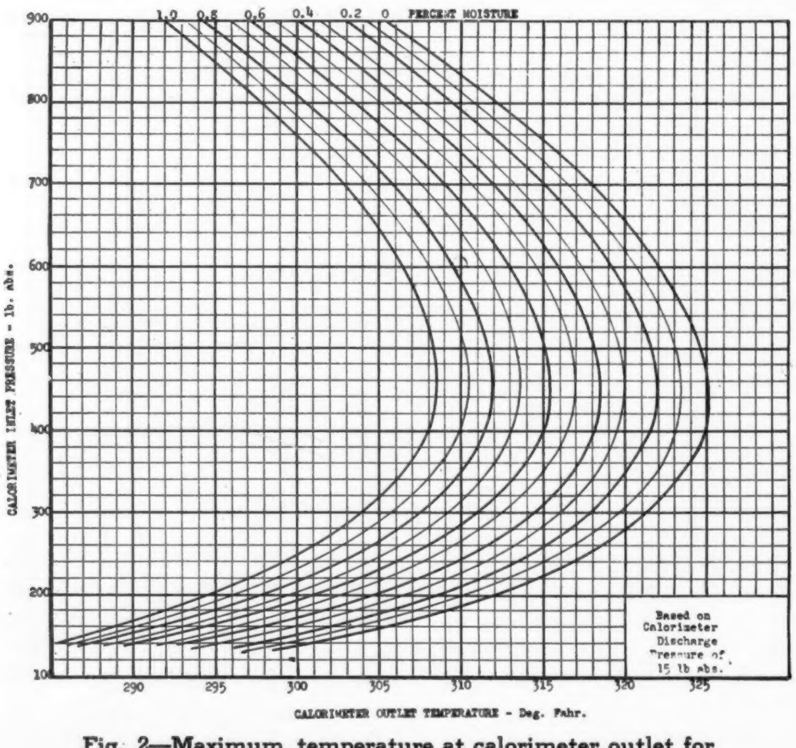


Fig. 2—Maximum temperature at calorimeter outlet for different inlet pressures and moisture up to one per cent

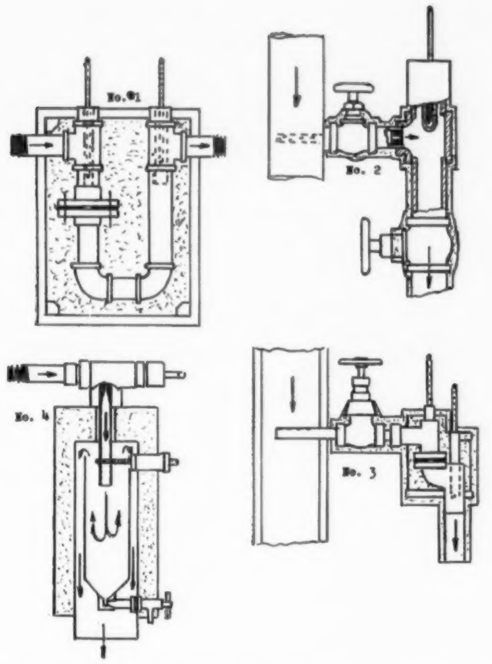


Fig. 4—Common types of throttling calorimeters

do not conform with all of the basic requirements given above is indicated by an allowance for a 10 to 12 deg normal correction in the Power Test Code on Calorimeters. Present standards of testing do not tolerate such a large correction value and a calorimeter that has a normal correction greater than one or two degrees should not be considered satisfactory.

Analyzing the types given in Fig. 4, No. 1 is probably made of one-inch pipe and fittings and has a high heat capacity and radiation surface for the allowable steam flow. No. 2 likewise is low in flow capacity and high in radiation surface and also does not allow sufficient expansion before the outlet temperature is measured. No. 3 has a very small expansion space before the temperature is measured and the thermometer is not directly in the steam path. No. 4 has a feature of steam jacketing to reduce radiation losses. This type is used widely and is the best design of those shown.

All of these types except No. 4 use a thermometer well for the outlet temperature thermometer. Since this well is under no pressure in atmospheric exhaust calorimeters it has no value, increases the difficulty of measuring the correct temperature and tends to restrict the flow at the outlet. The outlet thermometer should be used bare and may be held in place by a rubber stopper.

In Fig. 5 is shown a design of calorimeter, made of pipe fittings, that has been used over a wide range of pressure and has given small radiation losses. No allowance is made for measuring the inlet steam temperature. Inlet steam pressure is measured by a separate test-pressure gage located near the point of sampling. If the inlet temperature is also desired for check purposes, a well may be added to the calorimeter between the valve and the orifice. The orifice is of the nipple type and connects the valve to the body of the calorimeter. All fittings before the orifice are suitable for boiler pressure. All fittings after the orifice are at low pressure and are standard.

The orifice may be a thin plate type retained between flanges but the simpler nipple type has less heat capacity and is easier to insulate. The expansion chamber after the orifice is four inches long and the cross-section is increased to 1 in. and then to $1^1/_2$ in. to reduce velocity and allow conversion of velocity into superheat.

There are two locations for measuring outlet temperature to allow for duplicate thermometers or for installation of a recorder with a calibrating thermometer. If the dual measurements are not desired, the return bend and second tee may be omitted and steam exhausted from the first $1^1/_2 \times 1^1/_2$-in. nipple.

When tests over an extended period are made, it is very desirable to use a recording temperature meter in conjunction with a calibrated thermometer. Such a meter should have a small-sized bulb and a recording range between 200 and 350 F. A recording potentiometer and a thermocouple can be used if preferred.

The calorimeter, inlet connection and valve should be well insulated with hair felt or canvas-covered magnesia block and should be protected from cold drafts from adjacent fans or windows.

The discharge from a calorimeter is noisy and should be invisible. A visible exhaust is due to condensation and may result from cold drafts or poor quality of the steam sample. If the steam is very wet and the calorimeter capacity is exceeded, the discharge will be very cloudy and unevaporated water may even be discharged.

Before installing the calorimeter orifice, the sample line and valve should be blown out. Dirt or mill scale will sometimes plug the orifice causing reduction in steam flow and relatively increase the radiation losses.

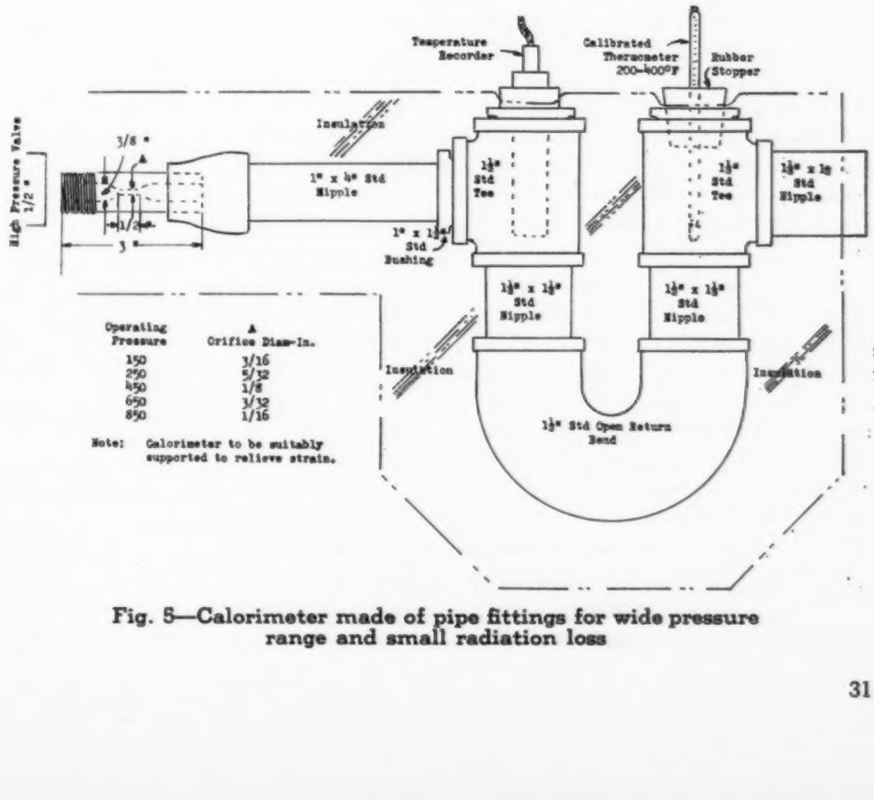


Fig. 5—Calorimeter made of pipe fittings for wide pressure range and small radiation loss

In order to obtain accurate results all of the available heat must be accounted for and it is necessary to reduce radiation losses to a negligible amount or else make suitable corrections for the heat lost. These corrections are involved in the determination of the so-called "normal" of the calorimeter.

If there is no moisture present in the steam sample and no radiation loss from the calorimeter, all of the available excess heat will be used for superheating the exhaust steam. The theoretical or maximum calorimeter outlet temperature under these conditions is the temperature of superheated steam at calorimeter discharge pressure for a total heat equal to that of saturated steam at the calorimeter inlet pressure and is found in approved superheated steam tables.

The observed maximum outlet temperature with dry steam may be expected to be 1 to 2 deg lower than the theoretical value because of radiation losses. The discrepancy between the observed and the theoretical temperatures constitutes a correction to be applied to all readings by that particular calorimeter. This correction covers not only radiation losses but also errors in temperature measurement that may be due to poor calibration or to stem correction of the thermometer. This correction is determined by reducing the steam output, water level and boiler water concentration to a point where a dry steam output can be assumed and checking the calorimeter performance under such conditions. The A.S.M.E. Code suggests a reduction in rating to about 20 per cent of normal but a better method is to reduce the rating slowly over a considerable period and range, and to record continuously the calorimeter temperature.

If the calorimeter indicates moisture in the steam at the higher ratings it may be expected that this carryover will decrease as the rate of steaming is reduced. When dry steaming is established, the calorimeter temperature should level off at a constant temperature.

It may be necessary to experiment to find the true normal. In some cases the rate of flow of steam in the line becomes too low and partial condensation occurs which registers as a moisture content that is higher than that obtained at higher ratings. The highest calorimeter temperature obtained during a slow and continuous reduction in rating may logically be taken as the true normal temperature.

The difficulty of establishing a normal is one important reason for designing the calorimeter so that its normal correction is not over one to two degrees. Failure to obtain a small and logical normal correction may be due to poor steam sampling, condensation in the steam line at low ratings or poor calorimeter design.

As stated above, it is customary to operate the calorimeter at atmospheric exhaust pressure. The actual exhaust pressure should be checked as it is involved in the calculations. This is easily done by removing the outlet thermometer and attaching a mercury U-tube at this point. The observed back pressure is converted to pounds absolute pressure and used in the following calculations for the exhaust pressure. If the back pressure does not exceed a half inch of mercury, the outlet pressure may be taken equal to 15 lb absolute without appreciable error.

Having established the calorimeter normal and its exhaust pressure the moisture in the steam can be calculated from the following calorimeter equation:

$$m = \frac{H_1 - H_2 - R}{L}$$

in which m = lb of moisture per lb sample
H_1 = total heat of saturated steam at inlet pressure
H_2 = total heat of superheated steam at exhaust pressure
L = latent heat of evaporation at inlet pressure
R = radiation loss in Btu per lb of the sample

This equation is derived from the following heat balance for the calorimeter:

Input to calorimeter $= mh + (1 - m)H_1$
where h = heat of the liquid at inlet pressure.
$(1 - m)$ = pounds of dry saturated steam per pound of sample

Output from calorimeter $= H_2 + R$, and equating the input and output we have

$$mh + H_1 - mH_1 = H_2 + R$$
$$mh - mH_1 = H_2 + R - H_1$$
$$m(h - H_1) = H_2 + R - H_1$$
$$m = \frac{H_2 + R - H_1}{h - H_1} = \frac{H_1 - H_2 - R}{H_1 - h}$$
$$m = \frac{H_1 - H_2 - R}{L}$$

The application of the equation to test data is best shown by an example.

Observed data

Calorimeter inlet pressure = 425 lb gage = 440 lb abs.
Calorimeter inlet temperature = 454 F
Calorimeter back pressure = 0.6 in. Hg
Calorimeter outlet temperature = 318 F
Calorimeter normal temperature = 323 F

Calorimeter outlet pressure
1 in. Hg = 0.49 lb pressure
0.6 in. Hg = 0.3 lb pressure
Outlet pressure = 14.7 + 0.3 = 15 lb abs

Theoretical dry steam temperature
Total heat saturated steam at 440 lb = 1204.6 Btu
Temperature of superheated steam at 15 lb and 1204.6 Btu = 325 F

Calorimeter normal correction
Temperature correction = 325 − 323 = 2 F
Btu correction = 2 × 0.475 = 0.95 Btu

Uncorrected test quality
Total heat saturated steam at 440 lb pressure = 1204.6
Latent heat at 440 lb pressure = 770.0
Total heat superheated steam at 15 lb and 318 F = 1201.3
Excess heat used for evaporating moisture = 1204.6 − 1201.3 = 3.3Btu
Pounds of moisture evaporated per lb of sample = 3.3/770 = 0.0045
Steam quality = 0.45 per cent moisture
99.55 per cent dry steam

Corrected test quality
Temperature correction = 318 + 2 = 320 F
Total heat superheated steam at 15 lb and 320 F = 1202.2
Excess heat used for evaporation = 1204.6 − 1202.2 = 2.4
Pounds of moisture evaporated = 2.4/770 = 0.0030
Btu correction = 1201.3 + 0.95 = 1202.3
Excess heat used for evaporation = 1204.6 − 1202.3 = 2.3
Pounds of moisture evaporated = 2.3/770 = 0.0030
Steam quality = 0.3 per cent moisture
99.7 per cent dry steam

Steam quality correction for normal = 0.45 − 0.30 = 0.15 per cent

Application of proper values to the calorimeter formula gives

$$m = \frac{1204.6 - 1201.3 - 0.95}{770.0} = 0.0030 \text{ lb}$$

Steam quality = 0.003 × 100 = 0.30 per cent

Moisture carryover from boilers may be classified under three types. First is a normal, relatively small

amount of finely divided droplets of water in suspension that escapes separation in the steam-drying equipment. This is the type on which guarantees are made and normally the amount will be less than 0.5 per cent. Second is moisture in the form of water films in a foam. This condition is not normal and results when the water conditioning is such that the steam bubbles do not readily break and the steam-liberating space fills completely with foam which is carried over as such. Third is moisture in form of solid slugs of water carried over during priming periods. This condition is also abnormal and usually is temporary and due to abnormal operating conditions such as high water levels in the drum or sudden changes in operating conditions.

Moisture carryover of the first type may be expected to vary with rating, concentration and water level but normally such changes should be gradual and not excessive. Sudden changes in quality and heavy carry-over indicate a change from normal carryover to abnormal carryover and abnormal conditions should be looked for.

Any boiler has a limited capacity for delivery of reasonably dry steam. When that capacity is exceeded, carryover of the second or third type occurs. This limit is dependent on boiler-water concentration, rating or sudden changes in rating, pressure or water level.

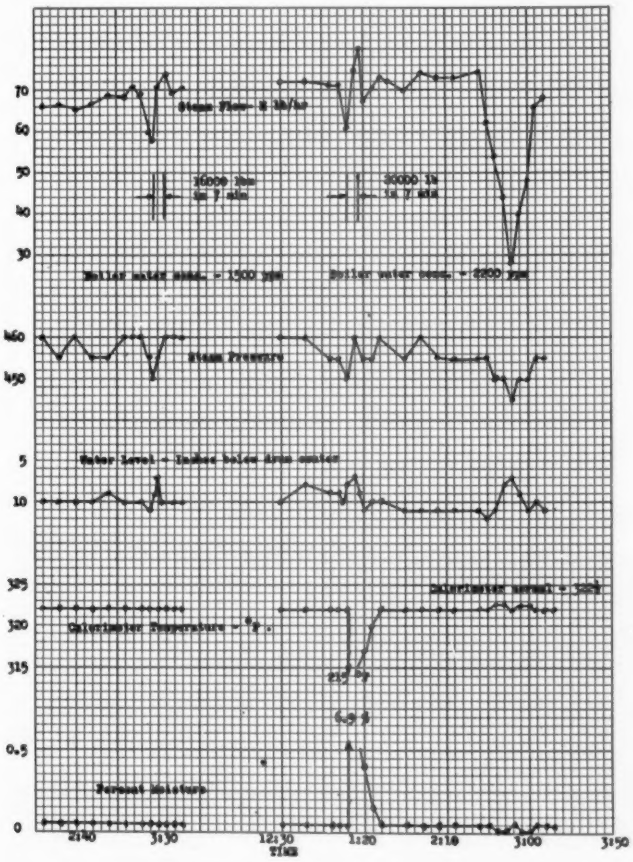


Fig. 6—Observed test data

In making calorimeter tests, data on general operating conditions should be collected in order to locate the cause of any excessive carryover that may occur. The following data are useful to this end.

1. Steam pressure.
2. Steam flow.
3. Superheated steam temperature.
4. Water level in drums.
5. Boiler-water concentrations.
6. Notes on (*a*) unusual operating conditions, (*b*) flue blowing periods, (*c*) periods of feeding chemicals to boiler and (*d*) rates of boiler blowdown.

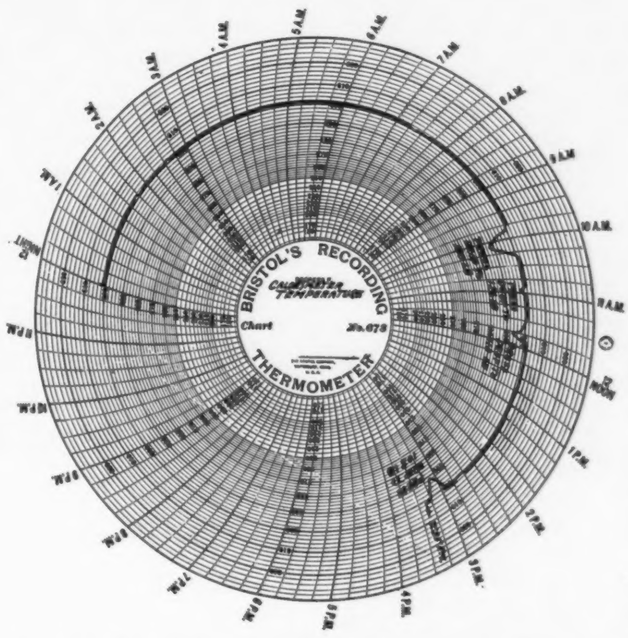


Fig. 7—Chart from calorimeter temperature recorder

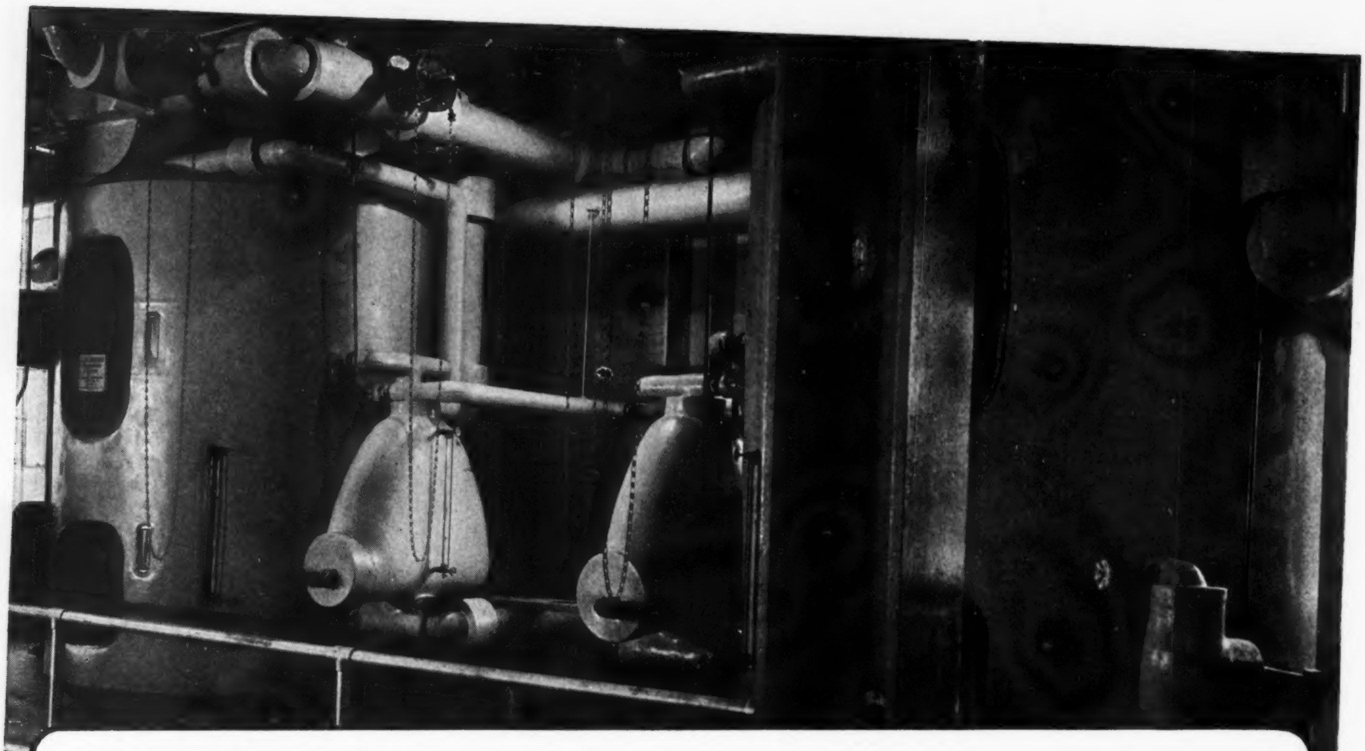
Sudden changes in operating conditions are particularly important in the interpretation of abnormal carryover. By observing all pertinent data over an extended period, during which the rating, water level and boiler-water concentration are varied over the desired range, and plotting such data on a single graph, the cause and correction of any sudden or excessive carryover are often made evident.

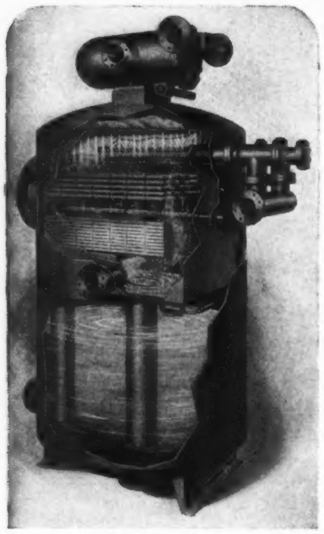
Correlated Data on Steam Quality

Fig. 6 shows typical correlated data of a steam quality test. With boiler-water concentration at 1500 ppm a swing in rating of 16,000 lb in seven minutes caused no change in steam quality. When, however, the concentration was increased to 2200 ppm, a change of rating of 20,000 lb in seven minutes caused priming. The calorimeter normal was determined by dropping the rating, the correction in this case amounting to approximately $1^1/_2$ F.

In some cases, changes in calorimeter temperature that seem to indicate changes in steam quality are actually due to changes in operating pressure. Fig. 7 is a chart of a calorimeter temperature recorder that shows four distinct dips indicative of carryover. Actually in all cases, the changes in temperatures were due to changes in operating pressure, the steam quality remaining constant.

In conclusion, it may be stated that although the throttling calorimeter has definite limitations in accuracy and application, it is still a very useful test instrument, especially when its use is correlated with other operating data.





DEAERATORS AT AMERICAN VISCOSE

- The new plant of the American Viscose Corporation at Front Royal, Va., is another of the great plants whose boilers are protected against corrosion by Cochrane Deaerators.

- The photograph above shows two vertical cylindrical cast-iron Cochrane Deaerators, each of 300,000 lbs. per hour capacity, deaerating boiler feed to the three high pressure boilers in this newest American Viscose plant.

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Metals at High Temperatures in Power Plants*

For piping and bolting, carbon-molybdenum steel is receiving the greatest amount of attention, hence its properties, the control of grain size, heat-treating practice, creep rates and kindred matters are discussed. Based on two specifications for bolting material issued by the A.S.T.M., the authors discuss the selection of properties, heat-treatment practice and the several causes of failure in this type of material.

By A. E. WHITE[1] and C. L. CLARK[2]

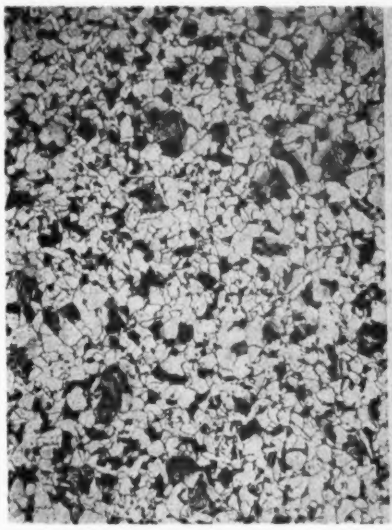

Fig. 1—Photomicrograph of carbon-molybdenum steel having low creep rate

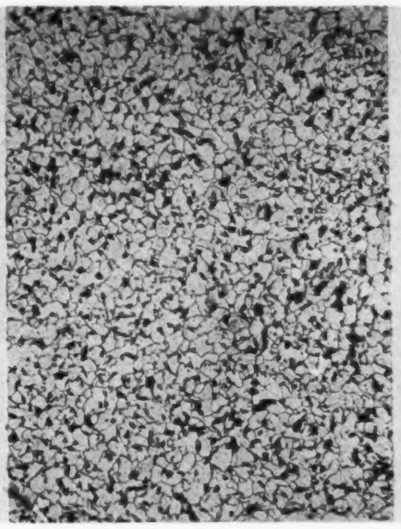

Fig. 2—Photomicrograph of carbon-molybdenum steel having medium creep rate

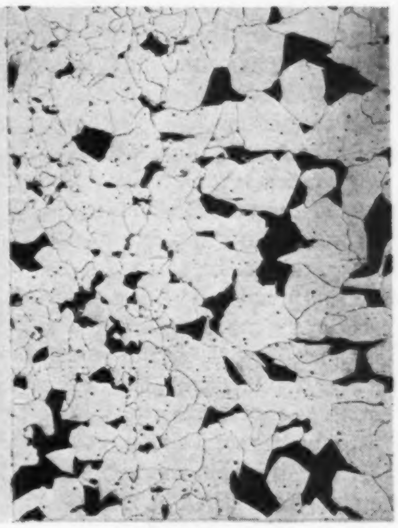

Fig. 3—Photomicrograph of carbon-molybdenum steel having high creep rate

THE alloy for pipe in the power-plant field, which today is receiving the greatest amount of attention, is carbon-molybdenum steel. It is used almost exclusively for high-temperature, high-pressure applications. It is by no means the only alloy which is available for this purpose, nor does it have as good high-temperature characteristics as some alloys which might be mentioned. But, for service temperatures up to 950 F and possibly up to 1000 F, it is the least expensive and the most nearly foolproof of all the alloys which are available. For these reasons, it will be the only pipe material which will be discussed.

One of the outstanding needs of those who use carbon-molybdenum steel for high-temperature purposes is a simple test which can be used to determine the high-temperature properties of the metal under examination. The standard creep test is too long, in that it is a test which requires 1000 hr for its completion. Other tests have been proposed, such as Hatfield's time-yield test, varying-rate tensile tests, stress-rupture tests and others. None of these has proved satisfactory at a temperature of 925 F for this steel, possibly because this temperature is in the strain-hardening range which introduces, in consequence, an uncontrolled variant.

Within the last two or three years, considerable attention has been focused on the metallographic method of determining acceptable high-temperature properties for plain-carbon and low-alloyed steels. The evidence to date seems to indicate that a steel, the normal tendency of which is to develop coarse grains, is preferable for high-temperature service to one in which the tendency is to develop fine grains. In other words, a steel which will

* Excerpts from paper at Semi-Annual Meeting of the A.S.M.E., Milwaukee, Wis., June 17 to 20, 1940.

[1] Consulting Engineer, and Director of Engineering Research, University of Michigan.

[2] Research Engineer, Department of Engineering Research, University of Michigan.

show a grain size of from 3 to 6 seems to have better high-temperature properties than a steel with a grain size of from 7 to 8. Yet, grain size in itself does not appear to be the only determining factor for any given composition. The nature and distribution of the carbides in the carbide-containing grains appears to be of considerable importance. For instance, when the carbide grains assume a Widmanstätten structure, that is, one in which precipitation occurs in cleavage planes, better high-temperature properties appear to be secured than when the carbides are in some other form.

Control of Grain Size

The control of grain size is a function of steel-melting practice and heat-treatment. In order that the steel-melting practice may be of an acceptable type, the steel should be either silicon- or silicon-aluminum-killed. It should not be killed with aluminum exclusively. Of course, any grade of steel can be made to acquire a coarse grain provided it be heated to a sufficiently high temperature, but a properly killed steel of the type desired for high-temperature service should show a 3 to 6 grain size if normalized or annealed from 1700 F. An improperly killed steel, on the other hand, if normalized or annealed from 1700 F, would show a fine grain.

In this connection, in some data which were obtained from an investigation sponsored by The Detroit Edison Company, creep rates at 925 F on various heats of carbon-molybdenum steel of substantially the same composition were found to range from 0.3 to 7.3 per cent per 100,000 hr, under the same stress.

The structures of the steels showing low, medium and high creep rates are given in Figs. 1, 2 and 3. The creep rate for the steel with the structure shown in Fig. 1 is 0.3 per cent per 100,000 hr. The grain size is from 5 to 7 and the carbide grains are of the Widmanstätten type. The creep rate for the steel shown in Fig. 2 is 2.3 per cent per 100,000 hr. The grain size is from 7 to 8. The creep rate for the steel shown in Fig. 3 is 7.3 per cent per 100,000 hr. The grain size in certain sections ranges from 2 to 3 and in other sections from 5 to 7. For the most part, the grain size is large. It is, however, decidedly non-uniform. Its structure is of a type known as duplex. The high creep rate is attributed at least in part to this type of structure.

Too broad generalities must not be drawn from any of these findings. Our knowledge with regard to what causes differences in high-temperature properties and how to interpret the findings of microstructures is still in an experimental stage and, before definite conclusions can be drawn, much further work will be required.[3]

Although it is now possible to get a steel of what is assumed to be the desired type from a steel mill, the grain size and the constitutional characteristics of the crystals may not be of the most suitable type when the metal has been fabricated and installed in the power plant. Constitutional changes may be brought about during fabrication which may adversely affect the type of carbide crystals and grain size, with a resultant change in the high-temperature properties of the metal. For instance, when pipe is bent it is heated, and the temperatures employed for heating may affect grain size and the nature of the carbide crystals. If the temperature of the heating is too low, it may even result in bringing about an agglomeration of the carbides, which is known as spheroidization. Such changes as have occurred during the bending, other than those of spheroidization, can usually be taken care of by a normalizing or annealing of the metal from 1700 F. This operation is not always possible, because it is not always practicable to place a bent section of pipe into a normalizing furnace.

Also, sections of pipe are now being assembled to an increasing degree by welding. This, of course, introduces temperature gradients ranging all the way from the temperature of molten steel to room temperature. The welding operation also introduces changes in chemical composition. Where it is possible to normalize or anneal the material thus welded in a furnace from 1700 F, most of the objections with regard to changes which have been brought about in the grain size and the grain structure are eliminated. Yet, it is not always possible to normalize or anneal the material from 1700 F. In most cases, it is possible only to stress-relieve it from 1200 F. This means, of course, that the grain size and the grain structure may be different from that which is preferred and, therefore, in certain sections, all of the precautions which have been taken to get preferred grain size and a preferred grain structure have gone for naught.

There is yet another matter which should, at least, be mentioned at this time. It relates to the tendency of of carbon-molybdenum steel, under certain conditions of time, stress and temperature, to become practically non-ductile. To be sure, most of the known cases to date, in which low ductility has been found, have been under conditions of a high or a relatively high stress. For instance, when a steel was held under a stress of 16,000 lb per sq in. at 1000 F, fracture occurred in but a few thousand hours, with elongations and reductions of area around 5 per cent for each.[4] This stress value is about three times the allowable stress value given by the A.S.M.E. Boiler Code for this kind of steel at this temperature. Some claim that, if the stress values are kept within those recommended by the Boiler Code, no brittleness will develop in the steel. Only time or a systematic investigation of this subject will tell whether or not this kind of steel will maintain normal ductility if conservative stress values are used.

Bolts in High-Temperature, High-Pressure Work

Although the use of bolts in high-temperature, high-pressure lines is rapidly declining, there are still places in which they are extensively used. Bolts are required for the assemblage of certain turbine parts. This use is most important. It is seldom, however, that bolts are used in modern piping systems, as welding has replaced most other types of pipe joints.

From time to time, failures with bolts are reported. These may be due to improper composition of material, improper heat-treatment, faulty material or abuse in installation.

The American Society for Testing Materials has issued two specifications for high-temperature alloy-steel bolting materials. One merely gives three classifications on the basis of tensile requirements with chemical limits for only sulphur and phosphorus. The other gives eleven

[3] Credit for the emphasis on the high-temperature properties of the metals on the basis of metallographic structure is given to S. H. Weaver, of the General Electric Co., and his associates.

[4] Courtesy of The Timken Steel and Tube Co.

types of steels with the full chemical composition of each type. Not all of these eleven types are suitable for all kinds of service. Also, there are many classes of steel which are not among the eleven types given but which have a place in this field.

In the selection of bolt stock, care must be exercised to see that the material is of a non-aging type. Further, since all bolt material shows lower impact values at some temperatures than at others, the bolt stock selected should be such as will have a good impact value at the temperature at which bolt is to be used.

Also, care must be exercised to see that the material will not develop low impact values in service. When found, these low values are a manifestation of aging that is produced by strains set up in the quenching operation when followed by a heating for the necessary time at moderately elevated temperatures or by an overstraining, followed, in turn, by a heating cycle.

As a rule, trialloy bolts are superior to dualloy bolts. In this connection, silicon is considered as an alloy when in quantities above 0.3 per cent. Many of the difficulties reported have been with dualloy bolting material.

Improper heat-treatment may be of two types; one a heat-treatment giving improper physical test values, and the other, one in which it is possible for quenching cracks and other defects to develop.

Heat-Treatment

In the matter of heat-treatment, there are two schools of thought; the one wishes high ductility in the finished bolt, even at a sacrifice of strength, while the other seeks high strength, even at a sacrifice of ductility. Personally, the authors favor bolts with high-strength characteristics as these resist distortion when they are tightened to a greater extent than the low-strength bolts.

In the realm of poor heat-treatment practice, the mass charging and mass quenching of bolt stock and bolts may yet be found in some plants. To the principle of mass production there is no objection but, when carried out without due regard to each individual piece, there is decided objection. Each piece must be heated and quenched uniformly, otherwise, nonuniformity of stock results. Some of the consequences of nonuniformity in heat-treatment are quenching cracks and too low or too high tensile properties.

Some of the failures which have taken place have been due to faulty material. This may mean an improper choice of material or defective stock. Defective stock may be due to lack of compliance with chemical and physical requirements, stock unduly filled with inclusions, stock with cracks due to faulty quenching or improper machining practice, or other causes of a similar character.

Finally, failures due to abuse in installation must not be forgotten. Less frequently than formerly, but nevertheless still sometimes occurring, undue strains are placed on the bolts during tightening-up operations. These lead to cracks which, in due time, result in bolt failures. Most companies now control the forces used in tightening the bolts so that excessive stresses are not set up, reducing, in consequence, the number of failures resulting from this cause.

The selection of proper bolting material is by no means as simple as may, at first glance, appear to be the case. Due regard must be given to room-temperature tensile properties, proper high-temperature properties, such as adequate creep strength, and acceptable impact values, both at room temperature and at the given elevated temperatures, with little, if any, drop in the values, even after long-continued service.

It is apparent from all that has been said that we are yet in the developmental stage so far as suitable alloys for pipes and bolts for high-temperature service are concerned. A master alloy which will resist oxidation and embrittlement, have high strength at elevated temperatures, irrespective of the nature of its constitutional structure, respond readily to working and welding, and, in addition, be reasonably inexpensive, has not as yet been found. Great progress in this direction has already been made and further progress is expected, as many metallurgists are today engaged in bringing forth new and improved alloys for high-temperature service.

Power Plant of the S.S. "America"

With the liner *America*, the largest merchant ship yet built in this country, now completed, interest among engineers centers around her power plant. This consists of six three-drum express-type boilers, having an aggregate output of 315,000 to 346,000 lb of steam per hour at 425 lb pressure and 725 F temperature from feedwater at 300 F. Each boiler is fired by six mechanical atomizing oil burners and is provided with a superheater, a submerged-coil type desuperheater in the steam drum and two horizontal tubular-type air heaters supplying combustion air at around 300 F. Six 22,000-cu ft per min. motor-driven forced-draft fans take air from the boiler room and deliver against 8 in. pressure to the air heaters. A part of this air passes through the double casings of the boilers.

In order to protect the decks from stack discharge, centrifugal-type dust catchers are installed in the uptakes.

The boilers are fed with distilled water and makeup is from evaporators. An independent system handles the returns from the heating system, laundry, fuel oil heaters, reciprocating pumps, etc. Stage heating is employed with two-point extraction from the main turbines and deaerating heaters on the first stage.

There are two main turbine sets having a combined output of 34,000 shaft horsepower and each consisting of a single-flow impulse-type high-pressure turbine operating at 3300 rpm, an intermediate-pressure turbine of the single-flow reaction type operating at 1500 rpm, and a 1500-rpm low-pressure turbine of the double-flow reaction type. The astern turbines, of the impulse type, are incorporated in the housings of the low-pressure turbines and represent a total of 19,500 shaft horsepower. Power is transmitted to the shafts through two sets of double-helical reduction gears. The high-pressure turbine of each set is located between the intermediate- and the low-pressure turbines and the condenser is placed below the latter. Instead of employing condenser circulating pumps, condensing water is supplied by means of scoops.

Auxiliary power for the ship is provided by two 600-kw turbine-generator sets running condensing.

NEW YORK LIFE INSURES AGAINST POWER FAILURE

SIXTY FEET UNDERGROUND in the sub basement of the New York Life Insurance Building a Hays Automatic Combustion Control System operates two oil burning boilers at a constant steam pressure and at the highest rate of combustion efficiency. In the Chief Engineer's report on the operation of his improved plant appearing in the January issue of National Engineer he says "The oil units respond so quickly and are so flexible it is not necessary to forecast changes in the load. The HAYS COMBUSTION CONTROL causes the burners to follow and satisfy these changes as they occur."

Changes in the steam header pressure cause an immediate response in the control system which operates to automatically regulate the oil flow and air supply to bring the pressure back to normal. Air and fuel are measured so as to maintain the most efficient and economical combustion which is indicated on the panel board by the CO_2 recorders, draft gages and smoke alarm.

Hays Combustion Control Panel in the New York Life Building at 51 Madison Ave., New York.

New York Life Insurance Building

If desired automatic regulation may be cut off at any time and the boilers operated by hand from the panel through remote manual control. The Hays system is electrically operated by which reason there is practically no time lag between the necessity for correction and the corrections themselves.

Similarly in boiler rooms throughout the nation, Hays Automatic Combustion Control is lowering fuel costs and maintaining highest operating efficiency, in most cases paying its initial cost in a year or eighteen months of operation. Highly efficient engineer representatives are located in all large centers to work with power plant owners and operators in determining the necessity and cost of instrumentation and control. You are invited to use this service without obligation. Write to 920 Eighth Avenue for further information.

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Application of Fan Laws

By G. A. HENDRICKSON

Dean of Engineering, Lawrence Institute of Technology

Performance data for a fan or pump under given operating conditions are readily calculated, by the use of fan laws, from the manufacturer's constant-speed characteristic curve or equivalent data. Applications of these laws are ordinary problems in ratio and proportion. The problems encountered in practice, however, are not always simple. Calculations in which two or even three laws are involved simultaneously are common, and the choice of the combination best suited to a particular situation may be baffling unless some systematic approach is adopted. It is proposed here to present a tabular statement of the fan laws from which the combination applying to a particular problem can be chosen mechanically. The use of this table is illustrated by an example.

IN DEALING with fans it is customary to assume that the gas handled is incompressible when the changes in pressure and in volume are relatively small. Fan theory derived from this assumption is not essentially different from pump theory; consequently, distinguishing terms are unnecessary. The word pump is used here as a general term to include both fans handling gases and pumps handling liquids.

Basis of the Fan Laws

The fan laws apply only to an operating condition in which the hydraulic efficiency of the pump remains constant as the rate of discharge varies. This efficiency may be defined as the ratio of the total head delivered by the pump to the sum of the total head delivered plus the head loss within the pump. That is,

$$e = h/(h + h_1) \quad \text{.................. (1)}$$

where

e = hydraulic efficiency
h = increase in total head through pump, in feet
h_1 = head loss by turbulence within pump, in feet

The nature of the variations in h and h_1 may be conveniently studied in a simple system such as that illustrated by Fig. 1. The head, h, represents simultaneously that delivered by the pump and that consumed in the system. It may, therefore, be concluded that if the loss within the pump and that in the system vary in the same proportion when the rate of circulation changes, the hydraulic efficiency of the pump, as indicated in Equation (1), remains constant. This condition is satisfied in any pump when the ratio of the velocity of flow to the peripheral velocity of the impeller is fixed.

The significance of this statement is illustrated in Fig. 2, in which velocity diagrams for a centrifugal pump operating at capacities of approximately 100, 75, 50 and 25 per cent of normal load are shown at a, b, c and d, respectively. In these diagrams u is the peripheral velocity of the impeller, w the velocity of flow relative to the impeller channels, and v the absolute velocity of flow.

When the ratio v/u is constant, as in these diagrams, the relation which the pump presents to the flowing fluid is shown at e. Here the vanes are considered stationary and are placed in a straight instead of a circular arrangement. The relative angle which the fluid stream makes with each wall with which it is in contact is the same as that shown in the velocity diagrams. This "stationary likeness" which neglects the function of the pump as centrifugal machine, compares the flow through the pump to the flow past obstructions in stationary channels in which the loss of head is proportional to the square of the velocity of flow.

The situation is the same in the actual impeller. When the fluid stream strikes each wall at a constant relative angle, the turbulent loss at that wall is proportional to the square of the velocity of flow. Since h, the loss of head in a given discharge system of the type considered, is proportional to the square of the velocity of flow in any condition, the ratio h_1/h is constant when the velocity ratio v/u is fixed, and under this condition the hydraulic efficiency does not change.

Each of the statements of proportionality and constant ratios as employed in the foregoing discussion of the necessary condition for a constant hydraulic efficiency,

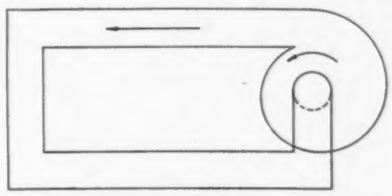


Fig. 1—Simple discharge system

is a simple fan law. For example, the condition that the velocity ratio v/u is constant, is equivalent to the statement that the discharge rate varies directly as the rotative speed. This is evident when it is noted that v can be expressed in terms of the discharge rate and u in terms of rotative speed.

Other statements can be similarly transformed, and all of the fan laws can be derived from them with the aid of a few commonly used physical laws. It is not the purpose here to make these derivations. The example given indicates that they arise from any operating condition in which the hydraulic efficiency remains constant.

Scope of the Fan Laws

At this point some attention to operation with varying hydraulic efficiency is desirable. The constant-speed characteristic curve is of particular interest. Fig. 3 shows such a curve, *kfaghij*, and the constant efficiency curve, *abcdo*, of the foregoing discussion is shown for comparison.

In this diagram the increase in pressure, *p*, between the inlet and the outlet of the pump is plotted against the rate of discharge *c*. Each plotted point bears the same reference letter as the corresponding velocity diagram in Fig. 2, where the effect of changing the velocity of flow, *v*, while the peripheral velocity *u* is maintained constant, is represented in *f*, *g*, *h* and *i*, and their accompanying "stationary likenesses" *f'*, *g'*, *h'* and *i'* are drawn for 125, 75, 50 and 25 per cent of normal discharge, respectively.

Each of these diagrams shows a different value of the velocity ratio v/u. The pump operating with v/u constant at any one of these values produces another curve in which the hydraulic efficiency is constant but different from that of the original curve *abcdo*. These curves are shown in the broken lines *og*, *oh* and *oi* of Fig. 3. Each point on the constant-speed characteristic *kfghij* has a

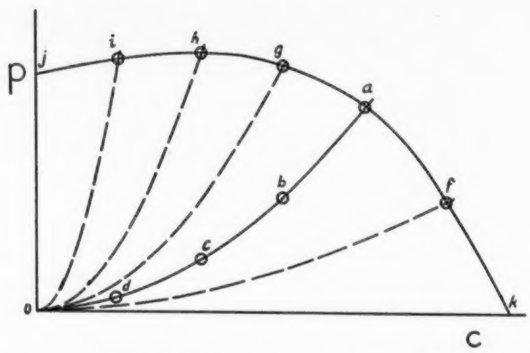


Fig. 3—Constant-speed characteristic and constant-efficiency curves

constant hydraulic efficiency curve passing through it to the origin *o*.

The variation in efficiency from one point to another, not on the same efficiency curve, cannot be predicted accurately, and it is impossible to use the fan laws between two such points. A statement of the pump performance must be available from the manufacturer at each rating used with the fan laws. These data are usually given in the form of a constant-speed characteris-

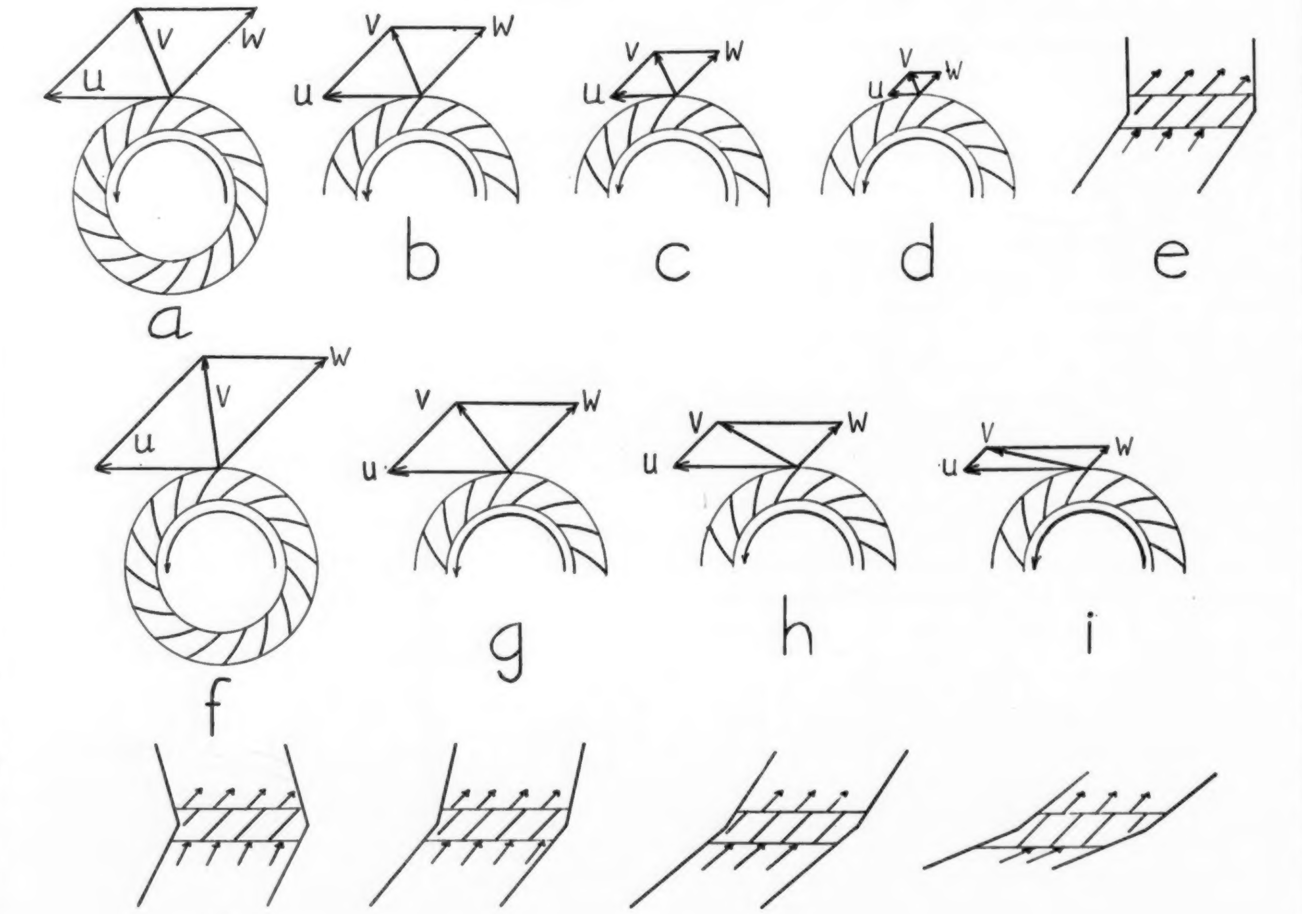


Fig. 2—Pump velocity diagrams and flow characteristics

tic curve or a table of performances at different ratings for some standard fluid density and rated speed. This specification ordinarily differs from the operator's requirements, and corrections must be made through the fan laws.

Fan Laws

Numerous fan laws may be written. Ten or twenty of the most common are usually stated, and corrections are made through the necessary combination chosen from those given. This procedure is greatly simplified if it is observed that all corrections may be expressed in the general form

$$\frac{n_1}{n_2} = \left[\frac{m_1}{m_2}\right]^{x_{mn}} \left[\frac{d_1}{d_1}\right]^{y_{mn}} \left[\frac{s_1}{s_2}\right]^{z_{mn}} \qquad (2)$$

where

n = any item of operating data; capacity, c; head, h; etc., which must be corrected to a new operating condition
m = any other item of operating data for which both corrected and uncorrected values are known
d = fluid density, lb per cu ft
s = fan size, in a unit of length

Values of the exponents x_{mn}, y_{mn} and z_{mn} are chosen from the accompanying table. They are found in the row m and in the column n. Subscripts 1 and 2 indicate the two points at the same hydraulic efficiency between which correction is made.

For use with gases, when density is given in terms of temperature and pressure, Equation (2) may be written in a more convenient form,

$$\frac{n_2}{n_1} = \left[\frac{m_2}{m_1}\right]^{x_{mn}} \left[\frac{b_2}{b_1}\right]^{y_{mn}} \left[\frac{T_2}{T_1}\right]^{-y_{mn}} \left[\frac{s_2}{s_1}\right]^{z_{mn}} \qquad (3)$$

Over two hundred equations involving the different variables and combinations of the different variables in the tabulated statement of the fan laws, may be written directly in the form of Equation (3), and others may be derived. Some of these equations are simple physical laws, but the greater number are usable fan laws. The reader will find many possibilities.

Application of the Fan Laws

Use of the table is illustrated in the following example:

A No. 10 Sirocco fan with double inlet boxes and two-thirds double width delivers 98,830 cfm of air at 8 in. water static pressure and 478 rpm when the atmospheric pressure is 29.92 in. Hg and the temperature 70 F. What would be the discharge capacity at zero F with the same static pressure and the same atmospheric pressure?

A new capacity is asked. Use c for n in Equation (3). The static pressure is known at both points. Use p for m. Then write Equation (3) as,

$$\frac{c_2}{c_1} = \left[\frac{p_2}{p_1}\right]^{x_{pc}} \left[\frac{b_2}{b_1}\right]^{y_{pc}} \left[\frac{T_2}{T_1}\right]^{-y_{pc}} \left[\frac{s_2}{s_1}\right]^{z_{pc}} \qquad (4)$$

With exponents chosen from the table this equation becomes,

$$\frac{c_2}{c_1} = \left[\frac{p_2}{p_1}\right]^{1/2} \left[\frac{b_2}{b_1}\right]^{-1/2} \left[\frac{T_2}{T_1}\right]^{1/2} \left[\frac{s_2}{s_1}\right]^{2} \qquad (5)$$

Since $p_1 = p_2$, the ratio $(p_1/p_2)^{x_{pc}}$ is unity whatever the value of x_{pc}, and this exponent need not even be read from the table for this simple problem. A similar statement is true for the ratio s_2/s_1. Equation (4) would simply be written

$$c_2/c_1 = (T_2/T_1)^{1/2} \qquad (6)$$

Solution for c_2 gives

$$\begin{aligned} c_2 &= c_1\sqrt{T_2/T_1} \\ &= 98{,}830\sqrt{(460 + 0)/(460 + 70)} \\ &= 92{,}070 \text{ cfm} \end{aligned} \qquad (7)$$

The fan would deliver 92,070 cfm of air at zero F and 29.92 in. Hg against a static pressure increase of 8 in. H_2O.

TABULAR STATEMENT OF THE FAN LAWS

Values of the exponents n in the equation $\frac{n_2}{n_1} = \left[\frac{m_2}{m_1}\right]^{x_{mn}} \left[\frac{d_2}{d_1}\right]^{y_{mn}} \left[\frac{s_2}{s_1}\right]^{z_{mn}}$

Values of the exponent x_{mn} when d is constant, s is constant and x is variable

$m \rightarrow$ / $n \downarrow$	c	h	N	P	p	v	W
c	1	1/2	1	1/3	1/2	1	1
h	2	1	2	2/3	1	2	2
N	1	1/2	1	1/3	1/2	1	1
P	3	3/2	3	1	3/2	3	3
p	2	1	2	2/3	1	2	2
v	1	1/2	1	1/3	1/2	1	1
W	1	1/2	1	1/3	1/2	1	1

Values of the exponent y_{mn} when $d = kb/T$, s is constant and x is constant

$m \rightarrow$ / $n \downarrow$	c	h	N	P	p	v	W
c	0	0	0	−1/3	−1/2	0	−1
h	0	0	0	−2/3	−1	0	−2
N	0	0	0	−1/3	−1/2	0	−1
P	1	1	1	0	−1/2	1	−2
p	1	1	1	1/3	0	1	−1
v	0	0	0	−1/3	−1/2	0	−1
W	1	1	1	2/3	1/2	1	0

Values of the exponent z_{mn} when d is constant, $s = kvA$ and x is constant

$m \rightarrow$ / $n \downarrow$	c	h	N	P	p	v	W
c	0	2	3	4/3	2	2	0
h	−4	0	2	−10/3	0	0	−4
N	−3	−1	0	−3	−1	−1	−3
P	−4	2	5	0	2	2	−4
p	−4	0	2	−4/3	0	0	−4
v	−2	0	1	−2/3	0	0	−2
W	0	2	3	10/3	2	2	0

A = Outlet area of fan, sq ft
b = barometric pressure, in. Hg
c = discharge rate, cfm
d = fluid density, lb per cu ft
h = head, ft
N = rpm
P = horsepower
p = pressure increase between fan inlet and outlet, in. H_2O
s = fan size, in a unit of length
T = absolute temperature, F
v = velocity of flow, ft/sec
W = weight of fluid pumped, lb per hr

The rotative speed at this new capacity can now be computed by using N for n and p for m in Equation (3). This gives

$$\frac{N_2}{N_1} = \left[\frac{p_2}{p_1}\right]^{x_{pN}} \left[\frac{b_2}{b_1}\right]^{y_{pN}} \left[\frac{T_2}{T_1}\right]^{-y_{pN}} \left[\frac{s_2}{s_1}\right]^{z_{pN}} \qquad (8)$$

or

$$\frac{N_2}{N_1} = \left[\frac{p_2}{p_1}\right]^{1/2} \left[\frac{b_2}{b_1}\right]^{-1/2} \left[\frac{T_2}{T_1}\right]^{1/2} \left[\frac{s_2}{s_1}\right]^{-1} \qquad (9)$$

which becomes

$$\frac{N_2}{N_1} = \left[\frac{T_2}{T_1}\right]^{1/2} \qquad (10)$$

or

$$\begin{aligned} N_2 &= N_1\sqrt{T_2/T_1} \\ &= 478\sqrt{460/(460 + 70)} \\ &= 445.3 \text{ rpm} \end{aligned} \qquad (11)$$

If the original problem had asked for the capacity at a given static pressure p_2 different from 8 in. the new capacity would be found from Equation (5) if the ratio p_2/p_1 is substituted in that equation simultaneously with the ratio T_2/T_1.

In a similar manner the horsepower, weight of fluid pumped or any other item of performance data may be calculated from the manufacturer's data and the accompanying tabulated statement of the fan laws.

West Penn Tops Windsor

The Windsor Station at Power, W. Va., originally built in 1916 and the last extension added in 1922, is unique in that one-half of the plant is owned by the West Penn Power Company and the other half by The Ohio Power Company. The latter in 1937 topped its half by the installation of two 750,000-lb per hr steam generating units supplying steam at 1250 lb 925 F to a 60,000-kw turbine-generator which, in turn, exhausts at 235 lb to three 30,000-kw existing turbines.

A 60,000-kw topping turbine-generator was ordered by the West Penn Power Company at that time for its half of the station, but installation was deferred until May of this year when an order was placed with Combustion Engineering Company for two high-pressure steam generating units to serve this turbine-generator and the other three 30,000-kw existing low-pressure turbines, thus completing the topping program for the Windsor Station.

These steam generating units will be of the three-drum bent-tube type, designed for an output of 750,000 lb each and 1525 lb drum pressure, 925 F total steam temperature. They will be equipped with Elesco superheaters and economizers, Ljungstrom air preheaters, and each will be served by three C-E Raymond bowl mills. The furnaces will be of the continuous slag-drip type and will be tangentially fired from each corner.

The installation is scheduled for completion in 1941 and the station will then have a total installed capacity of 300,000 kw in high- and low-pressure units.

Engineering Societies Employment Service Incorporated

In order to satisfy certain legal requirements of the several states in which it operates, the employment service of the Four Founder Engineering Societies has been incorporated in New York State and its name changed to Engineering Societies Personnel Service, Inc. Over a period of 17 years, the Service through its offices in New York, Chicago and San Francisco has placed more than 20,000 engineers in private industry and, from 1930 to 1939, more than 10,000 on W.P.A. and other governmental projects.

At the first meeting of the corporation, held in the Engineering Societies Building in New York, Monday, June 10, plans were made to extend the operations of the Service to a greater degree. George T. Seabury, national secretary of the A.S.C.E., was elected president; C. E. Davies, secretary of the A.S.M.E., vice president; Otis E. Hovey, director of the Engineering Foundation, treasurer; and A. H. Meyer, secretary.

High-Speed Turbines

"Contrary to the preconceived ideas of many engineers, operating experience with both land and marine turbine units seems to indicate definitely that the newer high-speed turbines are more reliable and more economical than the older slower-speed machines," said Glenn B. Warren, turbine design engineer for the General Electric Company, speaking at the Semi-Annual Meeting of the American Society of Mechanical Engineers in Milwaukee on June 17. Mr. Warren pointed out that the trend to higher speeds has permitted a better use of materials in turbines and has lowered investment and operating costs under what they would have been had speeds not been increased.

The trend toward higher turbine speeds has been dictated by a number of factors. First, is the question of efficiency; and second, is the fact that it becomes very much easier to build reliable turbines for high pressures over of the load by newer and more efficient stations as they are installed from year to year, and to the relegating of the older and less efficient plants to peak load and stand-by service.

The business depression of the early thirties reduced the number of new plants being installed, although the growth of the electrical load has continued. This resulted in an increasing use of many older and less efficient plants with a slowing up of the year-by-year reductions in the average coal consumption per kilowatt-hour. With the resumption of the power plant building program now under way, we should see still further reductions in the fuel consumption per unit of output. Mr. Warren stated that the progress of turbine design and construction since its inception has consisted largely in:

1. Building turbines and the attached generators of higher unit capacities to meet the growing demands for more power, and so permit reduction of plant investment per unit of output.

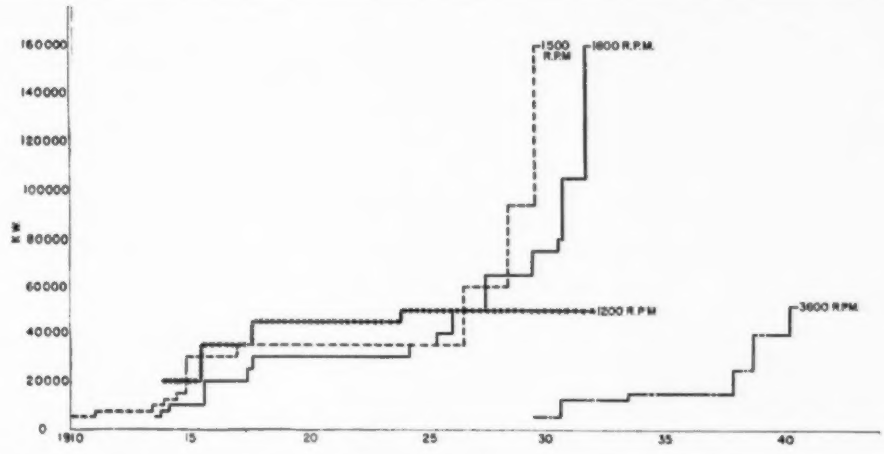


Fig. 1—Maximum capacity per unit (condensing)

and temperatures if the shells and rotors and interstage diaphragms are kept as small as possible. This can be done in higher-speed turbines.

Curves were presented summarizing the statistics of turbine progress which pointed to the following conclusions:

1. Although the largest possible unit at any given speed has steadily increased, and the large units have been effectively utilized in reducing operating and fixed costs on the large utility systems, the average size of unit sold in the United States (units above 10,000 kw considered) has increased but slowly.

2. Increases in maximum pressure and temperature have been followed by a steady rise in the average pressure and temperature for which machines have been sold, thus certifying to the fact that the pioneering done by a few power plant owners has been followed by an advance in the entire industry.

3. These facts are further attested by the almost continuous downward trend of the fuel consumption both for the best plants available and the somewhat similar trend for the average. Reduction in coal consumption of the best plants is due to the progress which has continued during the last few years, whereas the continual reduction in the average is due to the taking

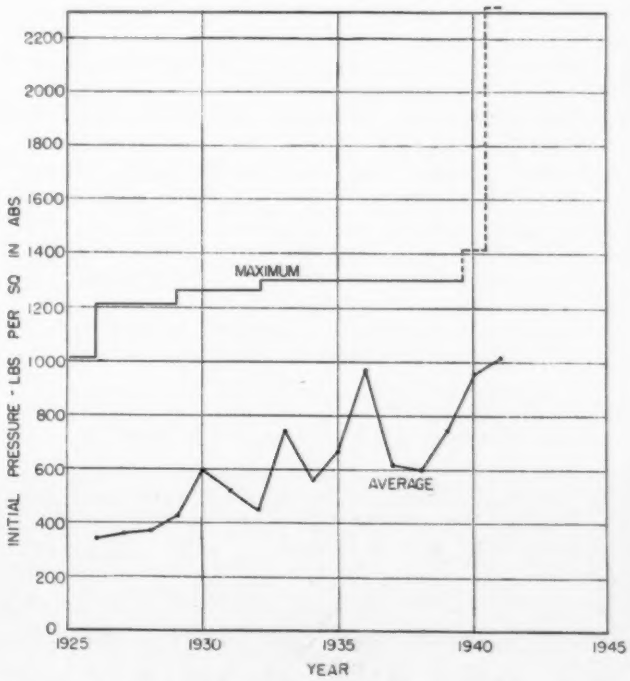


Fig. 2—Average and maximum steam pressures, 1925 to present

2. Continuously modifying and refining the design, manufacturing processes, and materials used in order to increase the reliability of operation and to reduce the outage time which has always been one of the chief concerns of operators of such equipment.

3. Utilizing higher initial steam pressure and temperature, coupled with improved heat cycles, such as regenerative feedwater heating, resuperheating, air preheating, etc., to decrease the fuel consumption per unit of output.

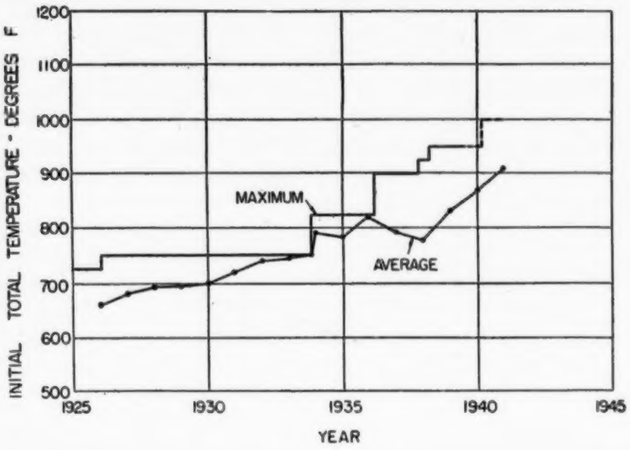


Fig. 3—Average and maximum steam temperatures, 1925 to present

4. Designing the turbine to utilize an ever-increasing proportion of the available energy in the steam cycle being used at the time.

Hydrogen-cooling has made possible larger 3600-rpm generators, and has within the past few years permitted building many turbines for 3600 rpm rather than 1800 rpm.

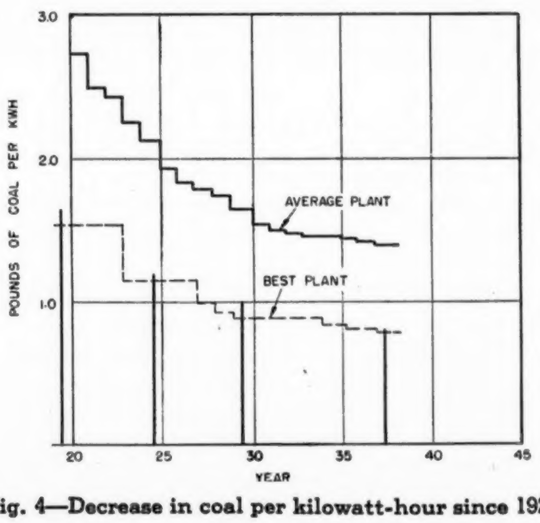


Fig. 4—Decrease in coal per kilowatt-hour since 1920

It was his opinion that turbine speeds for generator drive have probably reached a limit at 3600 rpm with 60-cycle power being generated. Although new electrical devices may change this situation, it does not appear to be in the immediate future, and new development should be stabilized at this speed, which will probably be an advantage for all concerned.

Oil Shale as Boiler Fuel

By H. R. TAUBE*

NECESSITY is the mother of invention. An interesting illustration of this adage has been furnished by Estonia, the little Baltic republic. After obtaining independence from Russia in 1920, she found herself in a precarious situation, cut off from former markets and sources of supply and confronted with the task of rebuilding her entire economic life. In the succeeding twenty years of independence Estonia has achieved impressive results.

Estonia is poor in natural resources, so that the fuel problem for her railroads and comparatively well-developed industries soon became of paramount importance. Her exports, mostly agricultural products and lumber, were needed to pay for urgent necessities and did not permit the importation of British or German coal to replace the only available domestic fuels, peat and wood. However, experiments undertaken by an enterprising engineer with oil shale, locally known as "burning stone," of which large but, supposedly, worthless deposits exist in Estonia, gave promising results. Today this fuel forms the basis of the country's heat economy and is used almost exclusively as industrial and locomotive fuel.

Furthermore, oil shale furnishes the raw material for the distillation of oil, a prosperous and quickly growing industry, producing not only crude oil, but also motor and lubricating oil, gasoline, asphalt, tar, etc. These products cover Estonia's domestic needs and have become one of the principal items of export trade.

The content of organic matter and ash of oil shale varies considerably in different strata of the deposits and locations. On the average, shale of good quality contains about 14 per cent moisture, 43 per cent organic matter and 48 per cent ash. Dry shale has the color of cocoa powder, burns with a long, smoky flame and has a heat value up to 6300 Btu per lb. In 1939 about one million tons of shale were used in the distillation of oil. The total output was 180,000 tons, or 18.76 per cent of the raw shale used, an increase over 1938 of 28.5 per cent.

The Estonian oil shale is similar in geological origin, chemical and physical properties to the oil shale of which vast deposits exist in Colorado, Utah and Wyoming, and which may, in the near future, become an important factor in this country's economic life, for the production of liquid fuel.

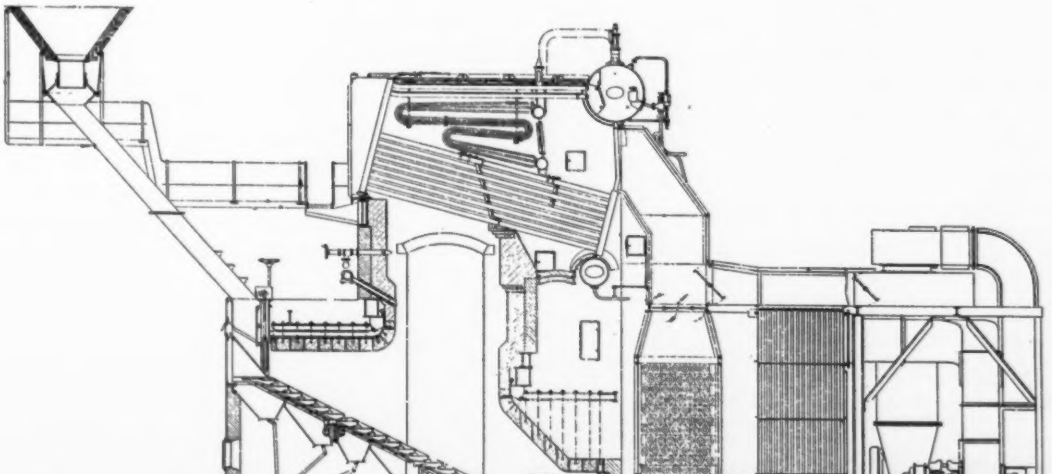
One of the first to introduce oil shale as boiler fuel was the municipal power plant in Tallinn, formerly Reval, the capitol of Estonia which has a population of 125,000 and is its industrial center. A pamphlet by A. Markson, manager of that plant, entitled "Oil Shale in an Electric Power Plant," gives interesting data concerning the design and operation of this power station. In 1938, it produced 30 million kwhr and the rising load curve is making necessary continuous expansion of the plant.

The adoption of oil shale as boiler fuel required radical changes in furnaces and firing equipment of the existing boilers and complete mechanization of fuel and ash handling, due to the high ash content. The plant, originally built for peat firing, could not be changed to permit the installation of furnaces of sufficient height required for complete combustion, and the efficiency of the old boilers, when fired with shale, never exceeded 70 per cent.

The oil shale as burned ranges in size from $^1/_2$ in. to $1^1/_2$ in. and contains, on the average, 13 to 18 per cent moisture, 38 to 44 per cent combustible and 37 to 45 per cent ash. Its cost, delivered to the boiler plant, is equivalent to about 20 cents per million Btu.

On the basis of tests and experience gathered over several years, a new boiler plant, designed especially for

* Engineering Department, Combustion Engineering Company, Inc., and member A.S.M.E.



Cross-section through unit burning oil shale at Tallinn Plant

oil shale, was erected in 1932. It is equipped with two British built B & W boilers of the conventional straight-tube, cross-drum type, with a heating surface of 5400 sq ft and a capacity of 33,000 to 38,500 lb per hr each at 355 lb pressure and 750 F total steam temperature.

In order to obtain a furnace of sufficient volume and enough space for ash removal in the basement, it was necessary to place the center line of the drums 42 ft 6 in. above the floor level. The furnaces are of the dutch-oven type with suspended arch, and are fired by forced-draft overfeed stokers with zoned air supply, as designed and built by the Ilmarine Engineering Co., Tallinn. The grate surface has been figured for burning low-grade shale with a heat value of only 4500 Btu per lb. For the maximum evaporation of 38,500 lb of steam per hour this would require the combustion of 14,330 lb of shale per hour with a grate surface of 345 sq ft, or a burning rate of 41.5 lb per sq ft per hr.

In Soviet Russia, where oil shale was first used as a boiler fuel and is still used in some districts, the firing rate and heat release per square feet of grate surface is much higher. This results in more compact and cheaper installations, though some of the savings in first cost thus realized are absorbed by more powerful fans and other auxiliary equipment. Taking further in consideration the greater wear and tear and an efficiency of only 65 to 70 per cent for these plants, Mr. Markson believes that the course followed in Estonia, where efficiencies of 83.6 per cent have been attained, is preferable to the Russian practice.

The gases leave the superheater at a temperature of 650 to 690 F and enter the economizer located directly behind the boiler. Here the feedwater is heated to 234 F, while the gas temperature drops to 410 to 446 F. At this temperature the gases flow through a tubular air heater having a heating surface of 6600 sq ft in which the air is preheated to 212 F.

A fan delivers this air to a chamber beneath the stoker, part of the preheated air being taken by an auxiliary fan and introduced into the furnace above the front arch through a row of nozzles. This overfire air mixes with the furnace gases, creating turbulence and making possible their complete combustion. From the air heater the flue gases at around 330 F pass through a centrifugal dust remover, of the Davidson type, which removes 85 per cent of the dust of a fineness such as would require more than one hour to sink a distance of 120 ft in still air. The result is that the gases leaving the 245-ft chimney are apparently dust-free, because the remaining dust particles are carried a long distance and scatter over a wide area.

The heat balance, at full capacity and for oil shale of good quality, is as follows:

Heat absorbed by boiler	63.27 per cent
Heat absorbed by superheater	12.57
Heat absorbed by economizer	7.77
Efficiency	83.61 per cent

The results obtained with this boiler plant were so satisfactory that last year an additional unit, similar to that described, but of approximately 100,000 lb per hr capacity was installed. Up to two-thirds full capacity shale alone is burned and above this shale oil is burned as a supplementary fuel. Oil is also used for starting.

A.C.I. Fuel Engineering Conference

WITH an attendance of nearly 400, Appalachian Coals, Inc., held its Twenty-Sixth Fuel Engineering Conference at the Shoreham Hotel, Washington, D. C., on June 21, under the chairmanship of J. E. Tobey. Following a welcoming address by C. C. Dickinson, President of the National Coal Association, several technical papers were presented at the morning and afternoon sessions.

E. G. Bailey, Vice President of Babcock & Wilcox Company, in a paper on "Progress in Use of Coal for Steam Generation," stated that nearly 90 per cent of the large units purchased during the past two years have been fired with pulverized coal in which field the most active developments in methods of firing and ash handling have taken place. Observing that coal analyses and the range of ash-fusing temperature are often not representative of the factors encountered in the furnace, he referred to the investigations of Gould and Brunjes and of Nicholls on the properties of coal ash which show the influence of certain fractions in the ash on its slagging properties. Furthermore, with coal having high iron content, heat releases up to 100,000 Btu per cu ft are possible in a primary furnace, but if the iron content is low heat releases up to 70,000 Btu represent about the limit. This is because iron acts as a flux and will lower the fluid temperature of the ash. Where a furnace is not forced beyond a heat absorption of 50,000 to 60,000 Btu per sq ft per hr the ash on the walls is usually in a dry state and can be handled satisfactorily.

Larger furnaces are desirable with coal than with oil in order to provide the necessary amount of heat-absorbing surface for reducing the gases to the proper exit furnace temperature. However, with coal it is less difficult to secure greater heat absorption in heat recovery surfaces of the unit, such as economizers or air preheaters, because deposits from oil are more likely to interfere with heat transfer in this region of the gas travel.

Discussing "Research and Progress," **Dean A. A. Potter,** of Purdue University, emphasized that under the incessant pressure of competition the coal industry can no longer afford to ignore research. It is needed to reduce production and distribution costs, to eliminate waste, to develop new uses for coal and to insure greater safety for mine workers. "Research and invention," he continued, "rather than governmental action, have in the past absorbed labor surplus and have elevated living standards by developing new materials, better processes, new industries, more economical methods of manufacture and additional markets."

T. W. Harris, Jr., Division Purchasing Agent of E. I. du Pont de Nemours & Co., whose paper on "Maximum Buying Under Minimum Prices" was read by W. H. Gehring, expressed the opinion that difficulties in purchasing coal will be materially increased upon the application of minimum prices as fixed by the Government. Buyers will expect more service from producers in lieu of increased prices and it will be impossible for the seller to make adjustment in prices to give the user the most economical coal for his particular case. Also, the absence of adjustment in prices for seasonal demand will likely necessitate stocking by the purchaser. He anticipated that quality will be one of the biggest factors under the new set-up and that it may be more economical to buy better coals than formerly. Finally, the fact that the present Act expires in April 1941, and may not be renewed, introduces uncertainties.

P. W. Swain, Editor of *Power*, in reviewing trends in industrial coal burning, touched upon fuel evaluation, coal preparation, fuel-burning equipment, plant modernization, efficiency and reliability, and called attention to the fact the private industrial steam plants consume approximately three times the coal burned by the electric utilities.

A paper by **P. Nicholls** and **W. T. Reid,** both of the U. S. Bureau of Mines, dealt with the properties of coal ash. Distinguishing between the fluid and the softening temperatures of ash, it was pointed out that two slags of the same fluid temperature may have quite different softening temperatures, or *vice versa*, and the nature of the slags may not be the same. Limitations of the cone method of determining such temperatures were discussed with reference to determinations in air and in a reducing atmosphere, and the development of the Bureau of Mines viscosity method was described.

At the luncheon brief addresses were made by Clyde E. Williams, Director of Battelle Memorial Institute, and by A. R. Mumford, Secretary of the Fuels Division, A.S.M.E.

A banquet was held in the evening with R. E. Howe, President of Appalachian Coals, Inc., acting as toastmaster, and addresses were given by Louis C. Johnson, Assistant Secretary of War, J. G. Stewart, Mayor of Cincinnati, and Dr. Harrison E. Howe, Editor of *Industrial and Engineering Chemistry*.

Secretary Johnson reviewed the problem of National Preparedness from the angle of industrial mobilization and emphasized the rôle of coal in such preparedness. "During the past year we again have noted that coal plays a vital rôle in the fate of nations," he said. "Germany made positive use of it to support an industrial war effort on a scale never before conceived by man. England vainly tried to use it in negative manner, in the style of 1914–18. She was still fighting the present war in terms of the last. She was placing reliance on the passive measures of denying coal and other materials to the enemy and to neutrals as a means of warfare. Today, France is on her knees. England has her back to the wall." "To us," he continued, "the lesson should be obvious. The enormous coal resources of this country must be capable of conversion promptly into sinews of war. Procrastination and delay must be side-tracked."

A.S.M.E. Nominations

Nominations for 1941 officers of The American Society of Mechanical Engineers were announced during the Semi-Annual Meeting of the Society in Milwaukee, June 17–20. The nominees as presented by the Committee are:

President

William A. Hanley, In Charge of Engineering,
Eli Lilly & Company,
Indianapolis, Indiana.

Vice Presidents

Dean Samuel B. Earle, School of Engineering,
Clemson Agricultural College,
Clemson, South Carolina.

Frank H. Prouty, Partner,
Prouty Bros. Engineering Co.,
Exchange Building,
Denver, Colorado.

Edwin B. Ricketts, Mechanical Engineer,
Consolidated Edison Co. of New York, Inc.,
New York, N. Y.

Managers

Prof. Huber O. Croft,
Head of Mechanical Engineering Department,
State University of Iowa,
Iowa City, Iowa.

Prof. Paul B. Eaton,
Charge of Mechanical Engineering Department,
Lafayette College,
Easton, Pa.

George E. Hulse, Chief Engineer,
Safety Car Heating & Lighting Co.,
New Haven, Conn.

REVIEW OF NEW BOOKS

Any of the books reviewed on these pages may be secured from Combustion Publishing Company, Inc., 200 Madison Ave., New York

Coal

By Elwood S. Moore

This is the second edition of the book which was originally published in 1922. It gives in detail the properties, methods of analysis, various classifications, geological formation, methods of extraction, uses of coal and location of fields. In addition to the known facts about its formation, the author presents the different theories offered by scientists as to origin of coal, the effects of climate on its formation, various stages in coalification and the effect of pressure on the vegetable matter which is the main constituent of coal.

One chapter is devoted to fossil flora of the coal-forming periods, another to structural features of coal seams and a third chapter contains a description of prospecting for coal and the valuation of coal lands.

The final chapters are devoted to the coal fields of the world and have many maps, diagrams, pictures and statistics, together with descriptions covering types of coal, depths of seams, amounts of known and probable reserves and other pertinent information. The book is well written and should be of value to anyone interested in petrology and chemistry of coal, classification, methods of extraction, preparation for market or utilization of this fuel. There are many illustrations and a complete index.

The book has 473 pages, 6×9. It sells for $6.

Design of Piping for Flexibility with Flex-Anal Charts

By E. A. Wert and S. Smith

The Blaw-Knox Company—Power Piping Division—has sponsored the publication of this book with the idea of placing in the hands of the experienced designer a series of charts and simple formulas which will enable him to obtain the stresses, forces, moments and deflections in piping designs of the one-plane, two-plane and three-plane structures in a much reduced time as against that required for doing this same work by the previous methods available.

Piping design for flexibility has always required a considerable amount of intricate calculation to obtain a satisfactory solution. The complexity of this problem increases considerably with the increase of temperatures and pressures now being demanded by the power industry and oil refineries.

The book may be obtained without charge by executives, engineers and others interested in piping design problems if they write their requests for the book on their business letterheads. To all others the book sells for $3. There are 91 pages, $8^1/_2 \times 11$, devoted to text, curves and photographs.

Dictionary of Metals and Their Alloys

Edited by F. J. Camm

This new book contains descriptions of all the known metals and many of the commercial metallic alloys together with their composition and characteristics. It has special sections devoted to plating, polishing, hardening and tempering, metal spraying, rustproofing, chemical coloring and useful tables.

The author, who is editor of a number of British magazines, believes it to be the first alphabetically arranged dictionary of metals and their alloys yet published. The book is adequately cross-referenced throughout and where necessary historical facts have been included.

There are 245 pages, size $5^1/_2 \times 8^3/_4$. The price is $3.

Electric Power Statistics

The Federal Power Commission has just issued a 61-page report entitled "Electric Power Statistics" which covers the production of electric energy, the installed capacity of electric generating plants, scheduled additions to generating capacity and construction expenditures for 1940, the consumption of fuel by electric generating plants for 1939 and previous years, the movement of electric energy across state lines and international boundaries and the production of electricity per capita as well as installed capacity per thousand population. Numerous tables and charts are included to depict this information and indicate trends.

To quote from some of these, the output of electricity for public use in the United States during 1939 was 130,336,050,000 kwhr, of which water power accounted for 33.8 per cent and of which privately owned utilities produced 88.3 per cent. The total installed capacity is given as 40,317,924 kw, made up of 28,046,948 kw of steam power, 11,415,165 kw of hydro and 855,811 kw of internal-combustion engines. The plant factors, computed by dividing the actual output for the year by the installed capacity multiplied by the hours during the year, figure 44.7 per cent for hydro plants as compared with 34.6 per cent for fuel burning plants and 38.3 per cent for privately owned plants as against 31.2 per cent for publicly owned plants. The scheduled additions to generating capacity for 1940 amount to 1,877,844 kw of which 1,332,635 kw are to be added by privately owned utilities and 545,209 kw by publicly owned utilities. The installed capacity per thousand of population is 307 kw and the consumption per capita is 994 kwhr per year.

In view of the rôle of power in the present preparedness program this report has particular significance.

The report may be obtained from the office of the Federal Power Commission, Washington, D. C., for 25 cents, and a "Supplement to Electric Power Statistics" containing detailed tables for each month of 1939 may be obtained for 25 cents additional.

Fuels and Their Utilization

By A. R. Carr and C. W. Selheimer

The text presents a comprehensive yet clear and simple discussion of the various fuels now in practical use. Here the whole subject of fuels is brought within the scope of one volume. The authors have made no attempt to go fully into thermodynamics, but have discussed both principles and applications of the means by which various fuels in current practice are utilized for the production of heat.

The text includes chapters covering combustion, classification of fuels, elimination of waste heat, heat balances, pyrometry and the like. It contains illustrative examples and laboratory experiments as well as problems following the various chapters.

There are 180 pages, $6^1/_4 \times 9^1/_4$. The price is $2.

Gas and Oil Engines

By P. S. Caldwell

This 125-page pocket-size volume is intended as a handbook on the running and maintenance of gas and oil engines, for the practical man and the student. It covers, among other things, engine details, governing, cooling, lubrication, starting and stopping, care, engine troubles, methods of testing and typical test results.

The price is $1.50.

Wage and Hour Manual, 1940 Edition

This manual published by The Bureau of National Affairs, Inc., is set up in three parts—The Fair Labor Standards Act, Wage and Hour Regulation Relating to Public Contracts and Wage and Hour Regulation by the States.

The part on the Fair Labor Standards Act is a reference source and guide to the Federal Wage-Hour Statute, topically arranged with regulations, official interpretations and determinations, decisions of the courts and authoritative answers to questions.

Part Two includes the Walsh-Healey Act, the Davis-Bacon Act, the Eight-Hour Law and the Anti-Kickback Act; and includes text of laws, rules and regulations, official rulings and interpretations and decisions of the courts.

The third part is a guide to state laws on wages and hours with digests of wage orders and decisions of the courts. This is followed by a complete topical index.

This manual sets forth the status of employees in many types of business and gives the rulings and interpretations on the pertinent questions asked about them. It should be very helpful to both employers and employees who may wish to know just what are the responsibilities and duties in regard to the various employer-employee relationships in any field.

The manual contains 648 pages, 6×9. The price is $5.

NEW EQUIPMENT

Blowoff Muffler

For separating sludge and water and dissipating pressure from blowoff lines in steam plants, a new type muffler has been developed by Sullivan Valve and Engineering Co. It is of the centrifugal type and is said to eliminate the disadvantages of a blowoff tank. By its use, steam can be vented through a sheet metal pipe and the water released through an open floor drain or direct to a sewer connection.

Two or more boilers may be connected to a single muffler without danger of back pressure being built up on an idle boiler, should a blowoff valve happen to be open. This muffler may be placed in any position within a boiler house or over a boiler setting, or on the roof, with a downspout leading to the sewer or sump, and a vent to the stack or outside of the building. The entire installation weighs only 50 lb.

Decarbonator for Degasification

A "Decarbonator" has been developed by the Cochrane Corporation primarily to meet the increasing demands of industrial and utility plants for trouble-free boiler feedwater. While complete degasification usually is accomplished during the heating of the feedwater, it may be desirable to remove such corrosive gases as carbon dioxide and hydrogen sulphide from the water in a cold condition without regard to other soluble gases such as oxygen.

The "decarbonator" is currently used to remove carbon dioxide from the effluent of carbonaceous hydrogen zeolite softeners and from water supplies treated with acid. Where soluble iron and carbon dioxide are present in the natural water supply, as they are in many parts of the country, it removes the dissolved carbon dioxide and by aeration oxidizes the soluble iron so that it may be filtered out and the water made fit for use.

Mercury Clutch

The new mercury clutch developed by Mercury Clutch Corp. permits a driving motor to gain speed before assuming load. Utilizing mercury to displace friction segments by centrifugal force, the clutch gradually picks up the load at full speed. Thus, smaller motors may be used when starting under load. Horsepower output per pound weight of the clutch is said to be greater than in comparable transmission units.

Only four principal parts comprise the clutch: the driving member or housing, the driven member or inner drum, the clutch segments and the mercury, the latter being introduced or removed through filler holes.

In operation, mercury displaces the clutch segments inward, where they engage the drum at the proper time and speed, a positive drive being effected when the speed of the housing and driven drum are synchronized. It is possible to predetermine the time or speed of engagement.

Overloads and other shocks are not transmitted through the clutch to the prime mover, the capacity of the clutch limiting the torque transmitted.

Motor Blower

A new "Motorblower" featuring quiet operation and low cost has recently been announced by the Ingersoll-Rand Company.

The unit can be installed on any floor since it requires no special foundation. A built-in blast gate is located in the blower discharge and flexible pipe connections are provided for connecting either intake or discharge to shop piping.

The maintenance of constant pressure over the entire volume range is stressed as being particularly advantageous in furnishing combustion air. It permits direct connection to several burners, any one or more of which can be turned off without affecting the operation of the rest. It is built in 72 sizes, pressures from $^1/_2$ to $2^1/_4$ lb, and volumes from 100 to 4500 cfm.

Pocket CO_2 Indicator

F. W. Dwyer Manufacturing Company announce an addition to its line of combustion testing instruments—the new No. 800 Dwyer Pocket CO_2 Indicator. It is inexpensive and said to be accurate and simple in operation. Among its features are the unbreakable transparent plaster construction, carrying case with built-in compartments for draft gage and thermometer, and self-closing valves eliminating any possibility of losing absorbent solution.

Receiver-Purifier

Centrifix Corporation has placed on the market a new type of receiver-purifier which is designed to solve the problem of the removal of oil in vapor form. It is claimed that expansion and contact capacity are sufficient to condense the vapor, and the design is such that after condensation the oil together with all other bothersome entrainment is removed by the Centrifix internal purifier.

Silica Removal

W. H. & L. D. Betz, Philadelphia, recognizing the importance of the silica problem in high-pressure high-temperature boiler operation has, after extensive research, developed the "Adsorption Process" for silica removal. This process which utilizes "Remosil" (especially prepared magnesium oxide) reduces soluble silica in any natural water to a fractional part per million while actually reducing rather than increasing the total solids in the water. Removal is effected in the hot-process softeners. No expensive additional equipment is required.

This process permits the addition of lime for softening operations when needed entirely independent of any magnesium requirements for silica control. It also makes unnecessary the use of other coagulants in softening operations and causes no increase in the requirements of lime and soda ash, but, rather, effects lower hardness and alkalinity of softened water than would otherwise be possible.

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